

DEPARTMENT OF THE INTERIOR

FRANKLIN K. LANE, SECRETARY

BUREAU OF MINES

VAN. H. MANNING, DIRECTOR

# THE DIESEL ENGINE

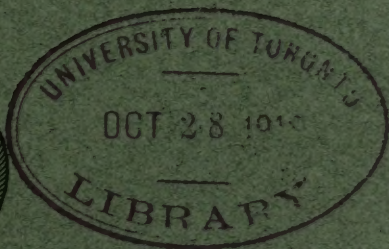
ITS FUELS AND ITS USES

BY

HERBERT HAAS

ENGINEERING

ENGINE STORAGE



WASHINGTON  
GOVERNMENT PRINTING OFFICE

1918





DEPARTMENT OF THE INTERIOR

FRANKLIN K. LANE, SECRETARY

BUREAU OF MINES

VAN. H. MANNING, DIRECTOR

# THE DIESEL ENGINE

## ITS FUELS AND ITS USES

BY

HERBERT HAAS



WASHINGTON  
GOVERNMENT PRINTING OFFICE

1918



The Bureau of Mines, in carrying out one of the provisions of its organic act—to disseminate information concerning investigations made—prints a limited free edition of each of its publications.

When this edition is exhausted copies may be obtained at cost price only through the Superintendent of Documents, Government Printing Office, Washington, D. C.

The Superintendent of Documents *is not an official of the Bureau of Mines*. His is an entirely separate office, and he should be addressed:

SUPERINTENDENT OF DOCUMENTS,  
*Government Printing Office,*  
*Washington, D. C.*

The general law under which publications are distributed prohibits the giving of more than one copy of a publication to one person. The price of this publication is 25 cents.

II

*First edition. June, 1918.*



# CONTENTS.

|                                                                        | Page. |
|------------------------------------------------------------------------|-------|
| Introduction.....                                                      | 1     |
| General characteristics and types of oil engines.....                  | 1     |
| Three general types of liquid-fuel engines.....                        | 2     |
| Explosion oil engines.....                                             | 3     |
| Engines having a four-stroke cycle.....                                | 3     |
| Engines having a two-stroke cycle.....                                 | 4     |
| Diesel engines.....                                                    | 6     |
| Engines having a four-stroke cycle.....                                | 6     |
| Engines having a two-stroke cycle.....                                 | 8     |
| Sabathé engines.....                                                   | 11    |
| Comparative advantages of four-stroke and of two-stroke cycles.....    | 12    |
| Essential qualities of engines of small power.....                     | 14    |
| Comparative economies of Diesel and of explosion oil engines.....      | 15    |
| Diesel's use of Carnot cycle.....                                      | 18    |
| Details of Diesel engines.....                                         | 19    |
| Vertical engines.....                                                  | 19    |
| Advantages of individual cylinder frames.....                          | 20    |
| Construction of cylinders and cylinder frames.....                     | 20    |
| Horizontal engines.....                                                | 22    |
| Construction of crank shaft.....                                       | 22    |
| Methods of lubrication.....                                            | 22    |
| Main bearings.....                                                     | 25    |
| Connecting rod.....                                                    | 25    |
| Pistons.....                                                           | 26    |
| Flywheel.....                                                          | 29    |
| Cylinder head.....                                                     | 31    |
| Air inlet and exhaust valves.....                                      | 35    |
| Fuel valves.....                                                       | 37    |
| Two functions of fuel valves.....                                      | 37    |
| Atomization of fuel.....                                               | 37    |
| Common type of atomizer.....                                           | 38    |
| Adjustment of pressure of injection air.....                           | 40    |
| Adjustment by hand.....                                                | 40    |
| Automatic adjusting devices.....                                       | 40    |
| Methods of fuel injection.....                                         | 41    |
| Hesselmann's fuel injector.....                                        | 41    |
| Needle valves with annular fuel passages.....                          | 42    |
| Use of open-nozzle fuel valve.....                                     | 43    |
| Equipment or methods permitting use of coal tar and coal-tar oils..... | 43    |
| Starting engine with gas oil.....                                      | 44    |
| Use of separate fuel pumps.....                                        | 45    |
| Retarding injection air and fuel.....                                  | 46    |
| Methods of removing fuel needle.....                                   | 46    |
| Methods of actuating valves.....                                       | 47    |
| System of rocking levers and cams.....                                 | 47    |
| Air-operated piston valves.....                                        | 49    |
| Details of valve-actuating devices.....                                | 50    |
| Valve-actuating mechanisms in horizontal engines.....                  | 52    |

## Details of Diesel engines—Continued.

|                                                                 | Page. |
|-----------------------------------------------------------------|-------|
| Fuel pumps.....                                                 | 53    |
| Pumps that supply fuel to open nozzles.....                     | 54    |
| Pumps designed to force fuel against high pressure.....         | 56    |
| Number of pumps required.....                                   | 57    |
| Air compressors and air receivers.....                          | 58    |
| General details of air compressors.....                         | 58    |
| Air receivers for air compressor.....                           | 60    |
| Regulation of air supply.....                                   | 61    |
| Material used in air pipes.....                                 | 62    |
| Source and volume of air used.....                              | 62    |
| Exhaust pipes.....                                              | 63    |
| Storage of fuel oil.....                                        | 63    |
| Cooling water.....                                              | 64    |
| Mechanical efficiency of different types of Diesel engines..... | 65    |
| Thermal efficiencies.....                                       | 66    |
| Volumetric efficiencies.....                                    | 67    |
| Effect of high altitudes on efficiency.....                     | 68    |
| Characteristics and uses of high-speed engines.....             | 69    |
| Diesel engines made in the United States.....                   | 70    |
| Allis-Chalmers Manufacturing Co., Milwaukee, Wis.....           | 70    |
| Busch-Sulzer Bros. Diesel Engine Co., St. Louis, Mo.....        | 72    |
| Dow Pump & Diesel Engine Co., Alameda, Cal.....                 | 72    |
| Fulton Iron Works, St. Louis, Mo.....                           | 72    |
| Fulton Manufacturing Co., Erie, Pa.....                         | 72    |
| Lyons Atlas Co., Indianapolis, Ind.....                         | 72    |
| McIntosh & Seymour Corporation, Auburn, N. Y.....               | 73    |
| National Transit Pump & Machine Co., Oil City, Pa.....          | 74    |
| New London Ship & Engine Co., New London, Conn.....             | 76    |
| Nordberg Manufacturing Co., Milwaukee, Wis.....                 | 76    |
| Southwark Foundry & Machinery Co., Philadelphia, Pa.....        | 76    |
| Worthington Pump & Machinery Corporation, New York, N. Y.....   | 76    |
| Liquid fuels for Diesel engines.....                            | 77    |
| Classification of fuels.....                                    | 77    |
| General characteristics of different fuels.....                 | 77    |
| Aliphatic hydrocarbons.....                                     | 79    |
| Petroleums.....                                                 | 79    |
| Desirable properties of petroleum fuel.....                     | 79    |
| Tests for suitability.....                                      | 79    |
| Fractionation or boiling-point tests.....                       | 80    |
| Effects of ash content.....                                     | 81    |
| Mechanical impurities.....                                      | 81    |
| Heating value.....                                              | 81    |
| Water content.....                                              | 82    |
| Coke residue.....                                               | 82    |
| Asphalt content.....                                            | 83    |
| Paraffin content.....                                           | 83    |
| Sulphur content.....                                            | 83    |
| Acidity.....                                                    | 84    |
| Elementary composition.....                                     | 84    |
| Viscosity.....                                                  | 84    |
| Flash point.....                                                | 84    |
| Burning point.....                                              | 84    |
| Specific gravity or density.....                                | 85    |



## Liquid fuels for Diesel engines—Continued.

|                                                               |       |
|---------------------------------------------------------------|-------|
| Aliphatic hydrocarbons—Continued.                             | Page. |
| Engine factors affecting combustion.....                      | 85    |
| Atomizer construction.....                                    | 86    |
| Size and shape of combustion space.....                       | 86    |
| Special methods used in burning heavy oils.....               | 86    |
| Two large classes for petroleum fuels.....                    | 87    |
| Specifications for fuel oils suitable for Diesel engines..... | 87    |
| Lignite-tar oils.....                                         | 89    |
| Aromatic hydrocarbons.....                                    | 89    |
| Coal-tar oils.....                                            | 89    |
| Specifications for tar oils.....                              | 90    |
| Coal tars.....                                                | 91    |
| Water-gas and oil-gas tars.....                               | 93    |
| Vegetable oils.....                                           | 93    |
| Arachis oil.....                                              | 93    |
| Palm oil.....                                                 | 93    |
| Composition and properties of typical liquid fuels.....       | 93    |
| Lubricating oils for Diesel engines.....                      | 94    |
| Bearing oil.....                                              | 94    |
| Cylinder oil.....                                             | 94    |
| Desirable properties of cylinder oil.....                     | 95    |
| Important advantages of Diesel engines.....                   | 95    |
| Low price of fuel.....                                        | 95    |
| Self-contained prime mover.....                               | 96    |
| Minimum fuel-storage space required.....                      | 96    |
| Full power quickly available.....                             | 96    |
| Reliability.....                                              | 97    |
| Comparative efficiencies of different prime movers.....       | 98    |
| Comparative cost data.....                                    | 98    |
| Cost of operation.....                                        | 98    |
| Cost of fuel.....                                             | 98    |
| Formulas for computing fuel cost.....                         | 102   |
| Significance of cost data.....                                | 102   |
| Effect of different factors on economy.....                   | 102   |
| Advantages of locomobiles.....                                | 104   |
| Limitations of suction gas plants.....                        | 106   |
| Conditions making other plants desirable.....                 | 106   |
| Factors affecting cost of generating power.....               | 107   |
| Summary of comparative advantages.....                        | 113   |
| Advantages of steam-turbine plants.....                       | 113   |
| Disadvantages of steam-turbine plants.....                    | 113   |
| Advantages of Diesel-engine plants.....                       | 113   |
| Disadvantages of Diesel-engine plants.....                    | 113   |
| Considerations governing choice of prime movers.....          | 114   |
| Special adaptations of Diesel engines.....                    | 116   |
| Examples of successful use of Diesel engines.....             | 119   |
| Pumping plants.....                                           | 119   |
| Ice manufacture.....                                          | 120   |
| Flour mills.....                                              | 120   |
| Central stations.....                                         | 121   |
| Locomotives.....                                              | 121   |
| Ship propulsion.....                                          | 122   |
| Selected bibliography.....                                    | 123   |

|                                                                                     | Page. |
|-------------------------------------------------------------------------------------|-------|
| Publications on petroleum technology.....                                           | 128   |
| Publications available for free distribution.....                                   | 128   |
| Publications that may be obtained only through the Superintendent of Documents..... | 129   |

## TABLES.

|                                                                                                              | Page. |
|--------------------------------------------------------------------------------------------------------------|-------|
| TABLE 1. Comparative economies of Diesel and of explosion oil engines.....                                   | 17    |
| 2. Limiting values of physical properties of typical coal-tar oil.....                                       | 90    |
| 3. Heat consumption and thermal efficiencies of different types of prime movers at continuous full load..... | 98    |
| 4. Comparative operating costs of different types of prime movers.....                                       | 98    |
| 5. Cost of various types of fuel.....                                                                        | 98    |

## ILLUSTRATIONS.

|                                                                                                                                                                                                  | Page. |
|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|-------|
| PLATE I. <i>A</i> , Bedplate of 120 brake-horsepower vertical engine; <i>B</i> , Construction of exhaust valve lever permitting easy removal of valve....                                        | 20    |
| II. <i>A</i> , Method of driving rotary lubricating oil pump in a vertical engine; <i>B</i> , Vertical governor shaft on a vertical engine driven through helical gears from the main shaft..... | 26    |
| III. Crank-effort diagrams for four-stroke engines with 1, 2, 3, 4, 6, 8, and 12 cylinders.....                                                                                                  | 28    |
| IV. Crank-effort diagrams for single-acting two-stroke engines with 1, 2, 3, 4, 6, and 8 cylinders.....                                                                                          | 30    |
| V. Crank-effort diagrams for double-acting two-stroke engines with 1, 2, 3, 4, and 6 cylinders.....                                                                                              | 31    |
| VI. <i>A</i> , Construction of fuel valve levers permitting easy removal of valve and needle; <i>B</i> , Diesel engine built by Allis-Chalmers Manufacturing Co.....                             | 52    |
| VII. Diesel engine built by Busch-Sulzer Bros. Diesel Engine Co., showing cam-shaft housing with housing covers open.....                                                                        | 53    |
| VIII. <i>A</i> , Busch-Sulzer 520 brake-horsepower Diesel engine; <i>B</i> , Busch-Sulzer 120 brake-horsepower Diesel engine.....                                                                | 70    |
| IX. <i>A</i> , Diesel engine built by Fulton Manufacturing Co.; <i>B</i> , Diesel engine built by Lyons-Atlas Co.....                                                                            | 71    |
| X. Cross section of Diesel engine built by Lyons-Atlas Co.....                                                                                                                                   | 72    |
| XI. <i>A</i> , Three McIntosh & Seymour 1,000 brake-horsepower Diesel engines; <i>B</i> , Three McIntosh & Seymour 500 brake-horsepower Diesel engines.....                                      | 72    |
| XII. <i>A</i> , Side view of Diesel engine built by National Transit Pump & Machine Co.; <i>B</i> , End view of Diesel engine built by National Transit Pump & Machine Co.....                   | 74    |
| XIII. Diesel engine built by Nordberg Manufacturing Co.....                                                                                                                                      | 75    |
| XIV. Vertical section of Diesel engine built by Southwark Foundry & Machinery Co.....                                                                                                            | 76    |
| XV. Diesel engine built by Southwark Foundry & Machinery Co.....                                                                                                                                 | 76    |
| XVI. Composition and properties of typical liquid fuels.....                                                                                                                                     | 94    |



|                                                                                                                                         | Page. |
|-----------------------------------------------------------------------------------------------------------------------------------------|-------|
| FIGURE 1. Explosion oil engine having a four-stroke cycle.....                                                                          | 3     |
| 2. Theoretical pressure-volume diagram of the operation of an explosion oil engine having a four-stroke cycle.....                      | 4     |
| 3. Explosion oil engine having a two-stroke cycle.....                                                                                  | 4     |
| 4. Theoretical pressure-volume diagram of an explosion oil engine having a two-stroke cycle.....                                        | 5     |
| 5. Diesel engine having a four-stroke cycle.....                                                                                        | 7     |
| 6. Diesel engine having a two-stroke cycle.....                                                                                         | 9     |
| 7. Working diagram of single-acting Diesel engine having a four-stroke cycle.....                                                       | 10    |
| 8. Working diagram of single-acting Diesel engine having a two-stroke cycle.....                                                        | 10    |
| 9. Indicator diagram of single-acting Diesel engine having a four-stroke cycle.....                                                     | 11    |
| 10. Indicator diagram of single-acting Diesel engine having a two-stroke cycle.....                                                     | 11    |
| 11. Pressure-volume diagram, Sabathé four-stroke cycle engine.....                                                                      | 12    |
| 12. Indicator diagrams of a Diesel engine having a four-stroke cycle..                                                                  | 13    |
| 13. Diagram explaining Carnot cycle.....                                                                                                | 18    |
| 14. System of lubrication especially adapted to high-speed engines...                                                                   | 23    |
| 15. Lubricating system in which oil is supplied through the bottom of each main bearing.....                                            | 24    |
| 16. Cylinder head of vertical Diesel engine.....                                                                                        | 32    |
| 17. Two-cycle engine with air admission through cylinder head, showing scavenging air valves.....                                       | 33    |
| 18. Two-stroke engine with two tiers of air valves.....                                                                                 | 34    |
| 19. Valves and valve fittings.....                                                                                                      | 36    |
| 20. Customary valve construction of Diesel engines.....                                                                                 | 36    |
| 21. Common type of atomizer.....                                                                                                        | 39    |
| 22. Patented throttling devices for Diesel engines.....                                                                                 | 41    |
| 23. Fuel injector developed by Hesselmann.....                                                                                          | 42    |
| 24. Open-nozzle fuel valve.....                                                                                                         | 44    |
| 25. Proportionate volumes of tar oil and of gas oil used in one power stroke of 50-horsepower cylinder at full load.....                | 45    |
| 26. Type of fuel-valve construction permitting easy removal of fuel needle or upper part of valve.....                                  | 46    |
| 27. Usual method of actuating valve gears of Diesel engines having four-stroke cycle.....                                               | 48    |
| 28. Movements possible by special mounting of rocking arms.....                                                                         | 49    |
| 29. Column stands supporting rocking arm lever and shaft.....                                                                           | 50    |
| 30. Elements of a gear for operating an air or an exhaust valve.....                                                                    | 51    |
| 31. View of cam for actuating fuel valve, showing slotted holes.....                                                                    | 51    |
| 32. Cross section of oil pump designed to supply oil to open nozzles without opposing air pressure.....                                 | 54    |
| 33. Cross section of oil pump designed to supply oil to open nozzle....                                                                 | 55    |
| 34. Essential features of oil pump designed to work against pressure of injection air and having constant-stroke plunger and push rod.. | 56    |
| 35. Distribution of power losses in two types of Diesel oil engines....                                                                 | 66    |
| 36. Typical performance curves representing engine efficiencies of 300-horsepower Diesel engines.....                                   | 68    |
| 37. End section of Diesel engine built by Allis-Chalmers Manufacturing Co.....                                                          | 70    |

|                                                                                                                                       | Page. |
|---------------------------------------------------------------------------------------------------------------------------------------|-------|
| FIGURE 38. Longitudinal section of Diesel engine built by Allis-Chalmers Manufacturing Co.....                                        | 71    |
| 39. Two sectional views of Diesel engine built by Busch-Sulzer Bros. Diesel Engine Co.....                                            | 73    |
| 40. Section through head of engine built by National Transit Pump & Machine Co.....                                                   | 74    |
| 41. Longitudinal section of engine built by National Transit Pump & Machine Co.....                                                   | 75    |
| 42. Plan and elevation of vertical three-cylinder Diesel engine with all auxiliaries.....                                             | 99    |
| 43. Plan of horizontal Diesel engine with all auxiliaries.....                                                                        | 100   |
| 44. Elevation of engine shown in figure 43.....                                                                                       | 101   |
| 45. Influence of size on efficiency of different prime movers at continuous full load.....                                            | 103   |
| 46. Average increases in fuel consumption by different prime movers at fractional loads.....                                          | 104   |
| 47. Heat balances of different kinds of engines.....                                                                                  | 105   |
| 48. Comparative costs of different items in the operation of a 100 horsepower Diesel engine at a mean 80 per cent load.....           | 108   |
| 49. Comparative costs of different items in the operation of a 200 horsepower Diesel engine operating at a mean 80 per cent load..... | 108   |
| 50. Operating costs for different engine-use periods reduced to corresponding horsepower-hour costs, 100-horsepower Diesel engine..   | 109   |
| 51. Operating costs for different engine-use periods reduced to corresponding horsepower-hour costs, 200-horsepower Diesel engine..   | 110   |
| 52. Comparative costs per unit of Diesel-engine equipment and of steam-turbine equipment for generating electricity.....              | 111   |
| 53. Curves showing cost of kilowatt-year (8,760 kilowatt-hours) at full load, as influenced by station capacity.....                  | 112   |
| 54. Curves showing cost of kilowatt-hour in mills at different station load factors.....                                              | 112   |
| 55. Comparative costs of operation of Diesel engine and of steam turbine installation for generating power.....                       | 114   |
| 56. Comparative operating efficiencies of Diesel engines and of steam turbines.....                                                   | 115   |
| 57. Influence of operating conditions on performance.....                                                                             | 116   |



# THE DIESEL ENGINE; ITS FUELS AND ITS USES.

---

By HERBERT HAAS.

---

## INTRODUCTION.

The Bureau of Mines is endeavoring to reduce waste and increase efficiency in the production, refining, and utilization of petroleum. During the last few years the demand for petroleum and its products has increased so much more rapidly than the production that greater efficiency in utilization is now a matter of most serious importance.

The Diesel engine, a comparatively new type of internal-combustion engine, is an important device for insuring more efficient utilization of petroleum and coal-tar products, for it consumes heavy liquid fuels such as can not be utilized in other types. Like the gas or the gasoline engine, it is revolutionary within its province. For some uses this engine is replacing gas and gasoline engines, but its most important duty will be to replace the wasteful consumption of fuel oil in steam engines. Not only will a wider use of the Diesel engine relieve the demand, but it will result in more power being obtained from the same consumption of fuel oils. In spite of these advantages, comparatively few Diesel engines are in use in the United States.

In this report the author discusses recent developments in the design and construction of the Diesel engine, the fuels suitable for burning in it, and the uses to which it is particularly adapted.

## GENERAL CHARACTERISTICS AND TYPES OF OIL ENGINES.

The term "oil engine," as generally used at the present time, is applied to internal-combustion engines that burn directly in the cylinder heavy liquid fuels of high boiling points, the fuel being injected into the compressed air shortly before or at the completion of the compression stroke. The distinguishing features of oil engines are that the fuel vapor is not absorbed by air before it is admitted to the cylinder, and that no inflammable mixture of vapor and air is compressed preceding its ignition. Oil engines compress air alone, and the heat of compression is used to ignite the fuel, which burns by consuming the oxygen of the air in the cylinder, the engine transforming the heat energy into work.

To facilitate and accelerate the burning of a liquid fuel it must either be vaporized, atomized, or intimately mixed with air immediately preceding its ignition.

Light, highly volatile liquid fuels, such as benzols, gasolines, alcohol, and distillates, offer no particular difficulties to vaporization; the air in its passage to the engine cylinder readily absorbs the fuel vapors and forms a combustible mixture, which is ignited electrically in the cylinder.

The process of charging the air with fuel vapors is called carburetion. The more volatile fuels, like gasoline, can be carbureted at ordinary atmospheric temperatures; that is, without previous heating of the air or the fuel. Liquid fuels with higher boiling points may require heating of the air or the fuel, or both, to bring about their evaporation and absorption by the air preceding combustion. In the heavy-oil engine the vaporizing of the fuel takes place inside of the engine. As a fuel with a high boiling point can not be evaporated at moderate temperatures, thorough mechanical division preceding ignition and combustion is necessary.

### THREE GENERAL TYPES OF LIQUID-FUEL ENGINES.

According to the means used for atomizing the liquid fuels and igniting them, there are two mechanically and thermodynamically distinct types of engines. In one type the entire fuel charge is sprayed against a highly heated surface in a chamber connected with the working cylinder. Contact with this highly heated surface gasifies the fuel, which is ignited and burns with explosion-like rapidity. Engines of this type are properly termed "explosion oil engines," or engines in which the fuel is burned at constant volume.

In engines of the other type the fuel to be converted is finely subdivided by air, and in this act of atomization is injected directly into the engine cylinder, where it is ignited automatically by the highly heated air in the cylinder. The combustion is not explosion-like, but is prolonged at constant pressure for the entire period during which the fuel is injected into the cylinder. This type of engine is universally known as the Diesel engine, being named after the late Rudolph Diesel, of Munich, Germany, its inventor. It is also termed a "constant-pressure oil engine."

There is a third general type of engine, combining features of the two types mentioned, in which the fuel is burned at both constant volume and constant pressure. Engines of this type are known as Sabathé engines.

According to whether an engine receives a working impulse every other revolution or each revolution, it is said to have a four-stroke cycle or a two-stroke cycle. Either type of engine may be single acting or double acting, such constructions being wholly mechanical and influencing in no manner the thermodynamic cycle of an engine.

All the three general types of oil engines enumerated may be built to work either on the four-stroke cycle or the two-stroke cycle and may be either single or double acting.



To summarize, differentiation must be made between the following types of oil engines:

1. Explosion oil engines.
  - (a) Engines having a four-stroke cycle.
  - (b) Engines having a two-stroke cycle.
2. Diesel engines.
  - (a) Engines having a four-stroke cycle.
  - (b) Engines having a two-stroke cycle.
3. Sabathé engines.
  - (a) Engines having a four-stroke cycle.
  - (b) Engines having a two-stroke cycle.

### EXPLOSION OIL ENGINES.

#### ENGINES HAVING A FOUR-STROKE CYCLE.

Figure 1 shows the elements of an explosion oil engine having a four-stroke cycle. During the first stroke of the piston downward, which creates a partial vacuum in the cylinder, atmospheric air is admitted through the air-admission valve *a* into the cylinder. On the return stroke of the piston valve *a* is closed, and the air in the cylinder is compressed. Before the piston is at the inner (upper) dead center the fuel charge is sprayed by the fuel pump *b* into the chamber *c*, known as a hot bulb or hot ball, which is kept at a dull, cherry-red heat. As the piston reaches its inner (upper) center, the fuel is burned with explosion-like rapidity, the temperature and pressure in the cylinder increase, and the piston receives a power impulse, the gaseous mixture expanding during the second downward stroke of the piston. On its return stroke the exhaust valve *d* is opened and the piston sweeps the gaseous products before it and through the valve *d*. The working cycle of four strokes is then repeated.

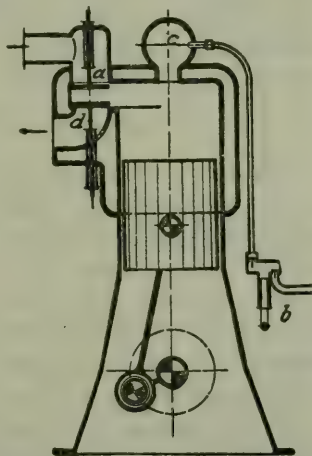


FIGURE 1.—Explosion oil engine having a four-stroke cycle. *a*, air valve; *b*, fuel pump; *c*, hot bulb; *d*, exhaust valve.

Before the engine is started the chamber or bulb *c* must be heated with a torch to a dark-red color, after which the successive explosions of the fuel oil will keep it hot. Inserted into this chamber in some engines is a thin-walled copper pipe against which the oil is sprayed. This small pipe can be heated quickly and readily absorbs heat from the chamber. It therefore tends to reduce the time required for starting the engine, which can thus start before the entire bulb is heated, and it also reduces missed ignitions after the

engine is operating. Heating of the bulb sufficiently to permit starting the engine requires 10 to 20 minutes. Compressed air, usually furnished by a small independent air compressor, operated by a gasoline engine, is used to start the engine.

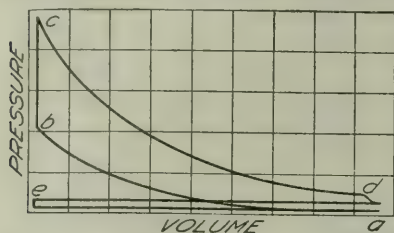


FIGURE 2.—Theoretical pressure-volume diagram of the operation of an explosion oil engine having a four-stroke cycle.

The line from  $c$  to  $d$  represents expansion, under which the piston moves forward, and at  $d$  the exhaust valve is opened, with the pressure dropping nearly to that of the atmosphere; the line from  $d$  to  $e$  represents the pressure-volume relation during the period in which the piston expels the products of combustion. The line from  $e$  to  $a$  shows a similar relation during the period in which the piston draws in a fresh supply of air, the pressure here represented being slightly below atmospheric pressure.

#### ENGINES HAVING A TWO-STROKE CYCLE.

Figure 3 illustrates the working of an explosion oil engine having a two-stroke cycle. Engines of this type have a closed crank case, provided on one side with large disk valves opening inwardly to the crank case and serving as air intakes. A large conduit connects the crank case with the cylinder, the conduit ports being so placed in the lower part of the cylinder that they are closed by the piston during the greater part of its travel; they are uncovered only a few degrees before it reaches its lower dead center.

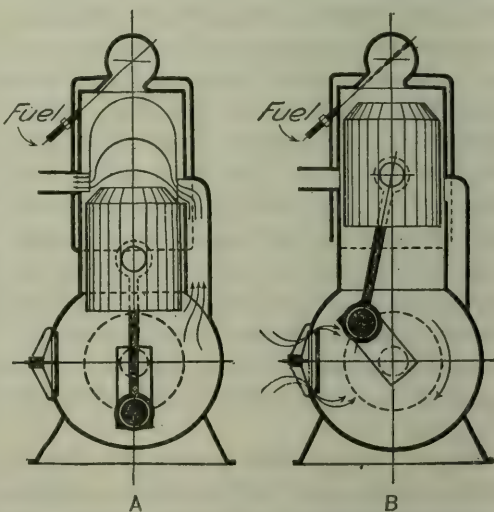


FIGURE 3.—Explosion oil engine having a two-stroke cycle. A, view when piston is at outer (lower) dead center; B, view when piston is approaching inner (upper) dead center.

Figure 3, A, shows the engine with a crank case full of air compressed by the previous downward travel of the piston, this internal pressure having closed the disk valve  $a$  leading to the atmosphere. The piston has traveled to its



outer (lower) dead center and uncovered the exhaust port *b* and the air port *c*. The pressure in the cylinder is higher than that of the atmosphere, so that the burned gases are swept out of the exhaust-gas port. The air port *c* admits fresh compressed air into the cylinder, sweeping it free of the products of combustion and refilling it with air needed for burning the next fuel charge. On its upward travel the piston covers the exhaust and air ports, compressing the air thus imprisoned in the cylinder. The air still in the crank case expands and its pressure falls below that of the atmosphere. Consequently the disk valve opens automatically and lets in a fresh supply of air (fig. 3, B). Just before the piston reaches its inner (upper) dead point a fuel charge is sprayed against the heated surface of the hot ball; the vaporized fuel is ignited and burned with a great increase of temperature and pressure by the time the piston has passed its inner dead point. The resulting impulse drives the piston downward, the two-stroke cycle being then repeated.

Figure 4 shows a pressure-volume diagram of such an engine. It is in principle the same as that of the engine having a four-stroke cycle, except that the exhausting of the gases, or scavenging, and the refilling of the cylinder with fresh air takes place along *dea*, *d* marking the point of opening of the exhaust ports several degrees before the outer dead point (at *e*) of the piston. Hence, only two strokes are required for a complete working cycle. These engines are started in the same manner as engines having a four-stroke cycle.

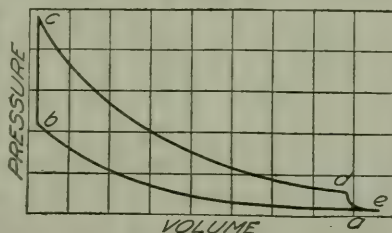


FIGURE 4.—Theoretical pressure-volume diagram of an explosion oil engine having a two-stroke cycle.

In explosion oil engines, the air is compressed in the cylinder to a pressure of 145 to 220 pounds per square inch and in the crank case to a pressure of 1 to 7 pounds per square inch. The explosion pressure varies from 270 to 470 pounds per square inch. The cylinder and the cylinder head are water-cooled, with the exception of the hot ball, which is kept at a dull red heat. The construction described is the one generally accepted and is to be preferred on account of its greater simplicity. Engines of this type are built both horizontally and vertically, the latter construction being preferred in multicylinder units.

The air for scavenging could also be furnished by constructing the working piston as a differential piston, traveling with the enlarged part in an air cylinder, or by an independent air pump. This design would result in a more complicated and expensive engine, hardly justified by its limitations.

## DIESEL ENGINES.

## ENGINES HAVING A FOUR-STROKE CYCLE.

Figure 5 shows the elements of a Diesel engine having a four-stroke cycle. Important adjuncts of the engine comprise a two-stage air compressor *a*, air-injection bottle *b*, compressed-air container *c*, and fuel-oil tank *d*.

The engine piston has started on its downward travel. The air-admission valve *e* is open to the atmosphere and lets air into the engine cylinder. On its return stroke the valve *e* and all other valves are closed, so that the piston compresses the air in the cylinder to a pressure of 450 to 500 pounds per square inch. As a result the air becomes highly heated, its temperature rising to about 1,000° F. when the highest pressure is reached. When the piston is at its upper dead center, the fuel-injection valve *f* is opened and liquid fuel in a volume proportioned to the load of the engine is forced into the cylinder, where, meeting the highly heated air, it automatically ignites and burns. When an automatically regulated supply of fuel has been delivered, valve *f* closes. Under the impulse of the expanding gases, the piston moves downward, transforming the heat energy of the fuel into work. Arrived at its lower dead center, the piston reverses its travel and begins a new upward stroke. The exhaust valve *g* being open, the piston sweeps the products of combustion before it, expelling them through this valve into the atmosphere, thus completing the four-stroke cycle. The cycle is repeated when air valve *e* opens again and the piston starts on its downward stroke.

For the sake of simplicity in describing the main features of the engine, no mention has so far been made of the important adjuncts of the fuel pump, the air compressor, and the fuel injector.

Pump *h* is under the influence of the engine governor, by which the volume of fuel delivered is proportioned to the load of the engine. The fuel oil stored in tank *d* flows by gravity to the pump, which delivers a measured quantity of oil to the fuel injector *f* during the suction stroke of the engine piston, while the needle valve of the injector is closed. The fuel injector is connected with the small air receiver *b*, in which air under a pressure of 700 to 900 pounds per square inch is stored, and air at this pressure always fills the valve chamber around the valve stem of the fuel injector.

Pump *h* in delivering its measured quantity of oil fuel to the injector must overcome the air pressure within the valve chamber. The pump is, therefore, designed to deliver the oil at a pressure of 100 to 200 pounds per square inch higher than the pressure of the air from the tank *b*, or at a pressure of 800 to 1,100 pounds per square inch. The fuel oil is delivered near the bottom of the fuel injector, immedi-



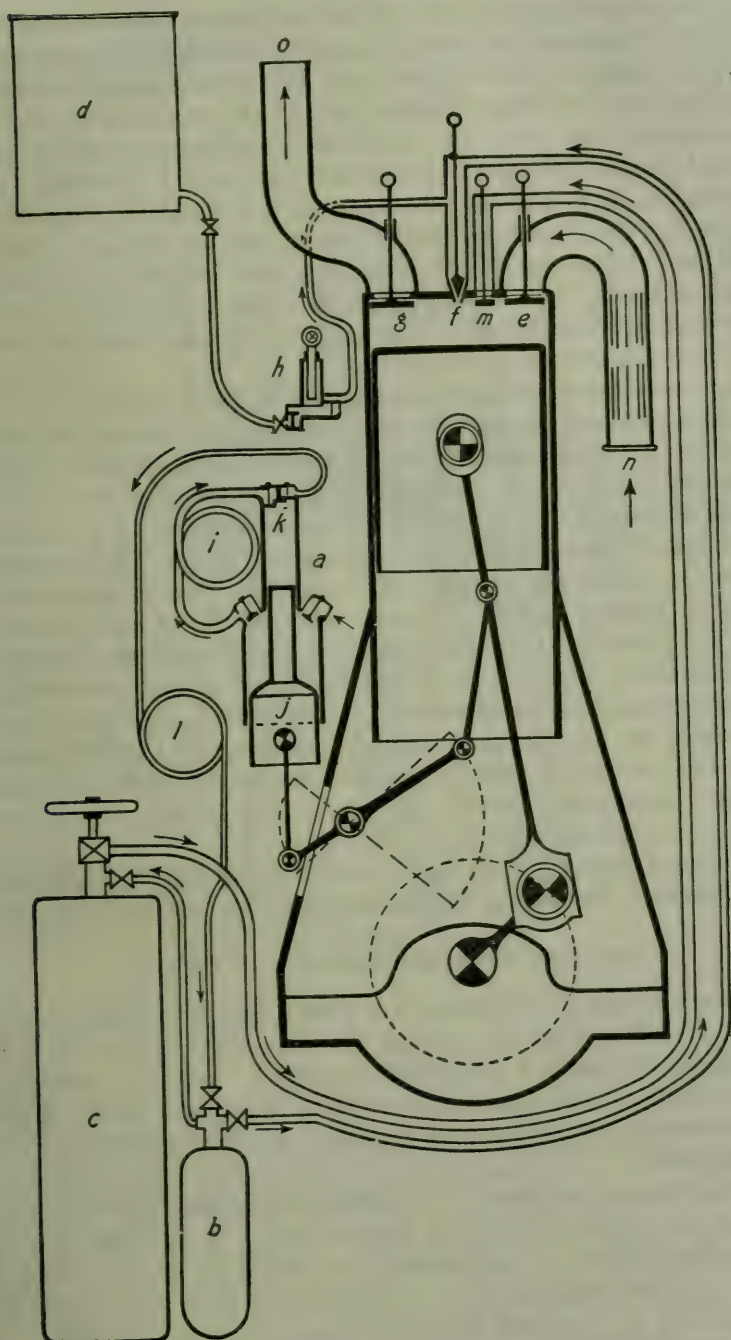


FIGURE 5.—Diesel engine having a four-stroke cycle. *a*, air compressor; *b*, injection air bottle; *c*, reserve air receiver; *d*, fuel tank; *e*, air (intake) valves; *f*, fuel-injection valve; *g*, exhaust valve; *h*, fuel pump; *i*, intercooler; *j*, low-pressure cylinder; *k*, high-pressure cylinder; *l*, aftercooler; *m*, starting valve; *n*, air intake; *o*, exhaust.

ately above the needle valve. When this needle valve is opened the injection air, which is at a pressure nearly double that of the compressed air in the engine cylinder at the completion of the compression stroke, atomizes the fuel oil and carries it into the engine cylinder. The injection air thus serves two important functions, namely, to inject the fuel into the cylinder and to subdivide it finely. The latter function is by far the more important, as on its efficiency greatly depends the success with which the heavy liquid fuels are burned. The fuel could be injected by direct pump pressure, but unless thoroughly atomized by highly compressed air it would burn only partly and after-ignition would result. This subject is treated more fully under the discussion of fuel injection.

The two-stage air compressor *a* furnishes the injection air; *i* is an intercooler to cool the air from the low-pressure cylinder *j* to atmospheric temperature before delivering it to the high-pressure cylinder *k*; *l* is an aftercooler to deliver cold air to tanks *b* and *c*. As previously mentioned, in the compressor the atmospheric air is brought to a pressure of 700 to 1,000 pounds per square inch. The air receiver *b* provides reserve air storage. Air is drawn from it whenever the engine is started and is delivered through the starting valve *m*, when the piston is at the upper dead center, in position to receive a power impulse, air being the power medium for starting the engine. After the engine has been started, tank *c* is recharged from tank *b*, by working the air compressor at its full capacity for 15 to 20 minutes. When the desired air pressure has been restored in tank *c*, all valves leading to and from the tank are closed. The air-inlet pipe is shown at *n* and the exhaust pipe at *o*.

Starting a Diesel engine from no load to full load requires one to three minutes, depending on the size and the construction of the engine.

#### ENGINES HAVING A TWO-STROKE CYCLE.

Figure 6 shows the elements of a Diesel engine with a two-stroke cycle. It differs from the engine just described in that it has a scavenging air pump *a* and exhaust ports *b*, arranged around the cylinder walls, covered and uncovered by the piston in its upward and downward travel, and two or more air-admission valves in the cylinder head.

The scavenging air pump *a* is double acting and its air delivery is controlled mechanically by a piston valve. The air admission to the engine cylinder is mechanically controlled by two or more air valves *c* in the cylinder head. The scavenging air is compressed to a pressure of 3 to 7 pounds per square inch.

The air is admitted into the working cylinder as soon as the exhaust ports have been uncovered by the piston; the cylinder, full of the gaseous products of combustion, is cleared by the scavenging air,



which sweeps out through the exhaust ports *b*, leaving the cylinder full of a new charge of fresh air. As the piston on its return stroke closes all cylinder valves and exhaust ports, the air is compressed in the same manner as in the engine having a four-stroke cycle, and

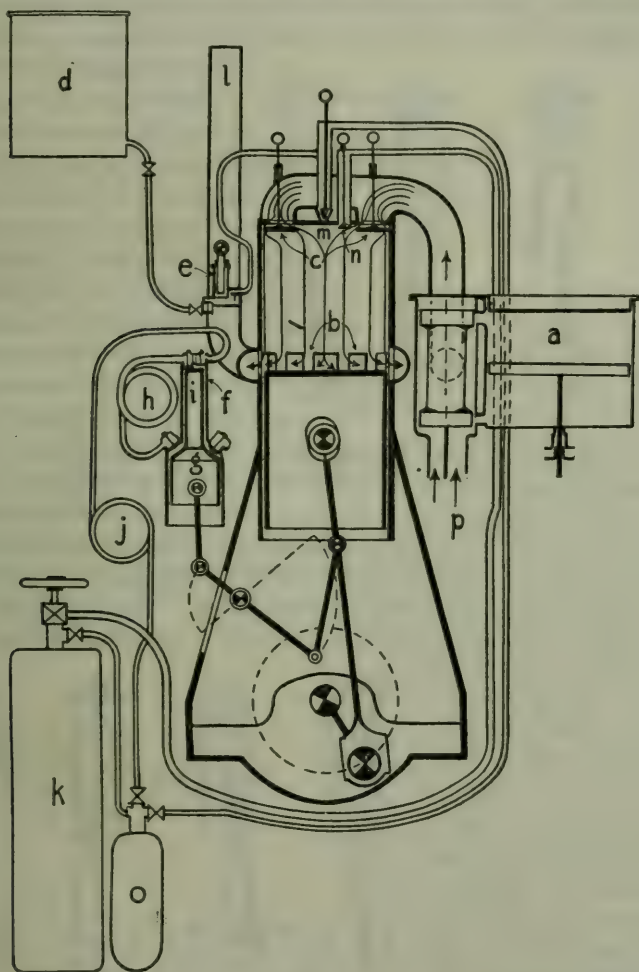


FIGURE 6.—Diesel engine having a two-stroke cycle. *a*, Pump for air for scavenging; *b*, exhaust ports; *c*, air valves; *d*, fuel tank; *e*, fuel pump; *f*, air compressor; *g*, low-pressure cylinder; *h*, intercooler; *i*, high-pressure cylinder; *j*, aftercooler; *k*, air receiver; *l*, exhaust pipe; *m*, fuel injection valve; *n*, starting air valve; *o*, injection air bottle; *p*, air inlet pipe.

the fuel charge is injected on the completion of the compression stroke.

During the next stroke the expanding gases give the piston its power impulse; with the opening of the exhaust ports and the discharge of the spent fuel gases, the working cycle is complete. By employing a scavenging air pump for cleaning the cylinder of its com-

bustion products and refilling it with fresh air, a working impulse to every revolution is obtained, whereas two complete revolutions to a working impulse are required in an engine having a four-stroke cycle.

In figure 6 the air-admission valves are shown in the cylinder head of the engine. In some later engines these scavenging air valves

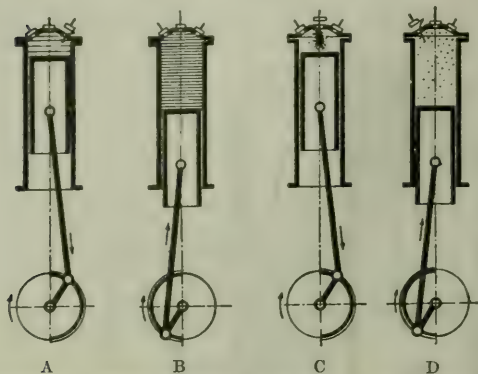


FIGURE 7.—Working diagram of single-acting Diesel engine having a four-stroke cycle. A, intake; B, compression; C, working stroke; D, exhaust.

have been placed in the cylinder wall, opposite the exhaust ports, somewhat similar to the placing shown in figure 3 for the explosion oil engine having a two-stroke cycle. The piston is shaped so as to direct the travel of the scavenging air in the cylinder and to arrange it in currents that sweep the cylinder clean of the products of combustion.

Thus the cylinder head is relieved of numerous air

valves, which, by reason of space limitations imposed by the cylinder head, have to be of restricted area, necessitating greatly increased air velocities to effect the desired scavenging. The placing of these valves in the cylinder head also results in a complicated head casting, the water spaces of which are narrow and restricted, so that effective cooling is made more difficult. By the modified construction above mentioned, the cylinder head has only starting and fuel injection valves. The placing of the air valves in the cylinder head has the advantage of insuring a more thorough scavenging, as the air, entering at the top of the cylinder, will more effectively sweep before it the burned gases.

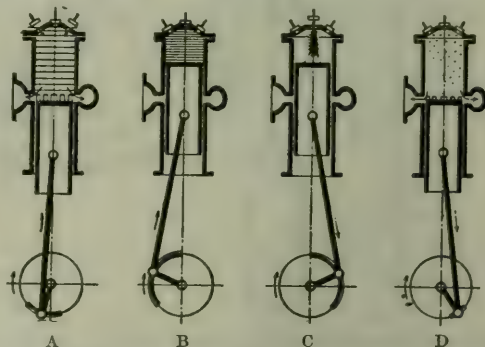


FIGURE 8.—Working diagram of single-acting Diesel engine having a two-stroke cycle. A, scavenging; B, compression; C, working stroke; D, exhaust.

The working cycles of the two engines are illustrated in figures 7 and 8; the corresponding indicator diagrams are shown in figures 9 and 10. The illustrations are taken from an article by Diesel.<sup>a</sup>

<sup>a</sup> Diesel, R., The Diesel oil engine: Engineering (London), vol. 93, 1912, p. 395.



## SABATHÉ ENGINES.

Sabathé engines differ from the Diesel engines described only in the method of admitting fuel and that of timing ignition, these being accomplished by a fuel-injection valve and a governor of a construction peculiar to these engines. A part of the fuel is admitted before the upper dead center of the piston is reached, and burns with constant volume. The remainder is admitted at the completion and return of the stroke, and burns at constant pressure. If the engine load diminishes, the fuel admitted during the second, or constant-pressure, period is diminished, stopping entirely during low load when the only fuel admitted is burned at constant volume.

The peculiarity of the working cycle of these engines is well illus-

trated in the pressure-volume diagram shown in figure 11. The burning of the fuel is accomplished at constant volume as represented by the line  $bc$ , and at constant pressure as represented by the line  $cc'$ . The fuel, of course, is not introduced intermittently at  $b$  and  $c$ , but is admitted continuously, the burning at constant volume and pressure being controlled by the action and timing of the needle valve, under the influence of the governor, at predetermined positions of the piston. Line  $cc'$  varies in magnitude with the load. At low load, line  $cc'$  disappears entirely; then the diagram becomes similar to that of an ordinary explosion oil engine. The diagram areas are proportionately decreased.

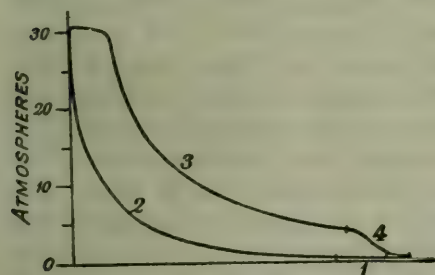


FIGURE 10.—Indicator diagram of single-acting Diesel engine having a two-stroke cycle. 1, scavenging; 2, compression; 3, expansion; 4, exhaust.

that it burns at constant volume. Line  $bc$  therefore inclines slightly toward the expansion line. The properties of the fuel, whether it burns readily or slowly, the temperature in the cylinder, and the means used for its ignition all influence the burning process. A part of the fuel usually does not burn until expansion has begun.

Likewise, in Diesel engine diagrams, there is no perfect constant-pressure line. With the admission of the fuel, there is a slight increase

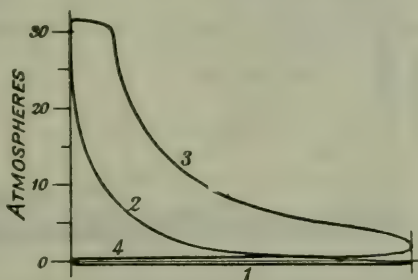


FIGURE 9.—Indicator diagram of single-acting Diesel engine having a four-stroke cycle. 1, intake; 2, compression; 3, expansion; 4, exhaust.

in pressure; there is also an after-burning of incompletely burned fuel when expansion has started, so that the initial and the terminal parts of the constant-pressure line are slightly curved.

In explosion oil engines the constant-volume line  $bc$  of figure 11 is shortened, and the diagram area is decreased, with a decrease in

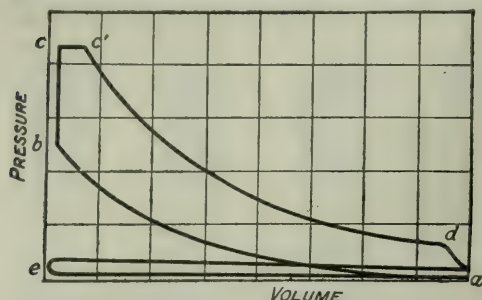


FIGURE 11.—Pressure-volume diagram, Sabathé four-stroke cycle engine.

the fuel supply during fractional loading of the engine. In the Diesel engine the constant-pressure line is shortened, and the diagram area decreased, with partial loads. In some two-stroke engines that increase the specific duty by increasing the scavenging air pressure; that is, by pumping a greater weight of air into the work-

#### COMPARATIVE ADVANTAGES OF FOUR-STROKE AND OF TWO-STROKE CYCLES.

As regards the relative theoretical advantages of the four-stroke and the two-stroke engines, the two-stroke engine appears to be superior. For a given cylinder size and speed, the specific duty of an engine having a two-stroke cycle is double that of an engine having a four-stroke cycle, for it receives a power impulse for every revolution as against one for every two revolutions of the engine having a four-stroke cycle. Its mechanical efficiency also would appear to be greater, as the engine with a four-stroke cycle has to make two strokes for expelling the products of combustion and for filling the cylinder with a fresh supply of air for burning the next fuel charge. The power for this work has to be drawn from the flywheel, in which it has to be stored again during the power stroke. As in addition the cyclic regularity of an engine with a four-stroke cycle but having the same power and speed is greatly less, a much heavier flywheel is required. The four-stroke engine is also much heavier per horsepower than the two-stroke engine.

In practice, however, these advantages are not so apparent. Engines having a two-stroke cycle require an auxiliary air pump for performing the work of scavenging and of refilling the working cylinder with fresh air, work that in the engine having a four-stroke cycle is performed by the engine directly in the working cylinder. The air



pump, proportioned to the engine it has to supply, is a large and heavy piece of machinery, requiring considerable power for its operation. To insure thorough scavenging of the working cylinder, a volume of fresh air greater than the volume of the cylinder must be supplied by the air pump. The pump is usually proportioned to supply one and three-tenths times the working cylinder volume of air. Even then sweeping of the working cylinder clean of the remnants of the products of combustion is difficult, and any gases remaining in the cylinder displace a proportionate amount of oxygen needed to maintain combustion, thus lowering the possible power output of the engine.

The heat exchange in an engine having a two-stroke cycle being more rapid than in one having a four-stroke cycle, the material of the former is more severely taxed, and it is not advisable to use as high mean effective pressures nor piston speeds as in engines having a four-stroke cycle. High mean effective pressure results in high terminal pressure at the time of exhausting, with a consequent loss in power. As the exhaust ports of two-stroke engines are uncovered by the piston

ton, and as exhausting of the gases must be rapid, the piston has to uncover these ports several degrees before it reaches its lower dead center. High mean effective pressure will therefore cause proportionately greater losses in an engine having a two-stroke cycle than in one having a four-stroke cycle. These features combine to diminish the theoretical advantages that engines having a two-stroke cycle appear to possess over engines with the simpler four-stroke cycle.

To avoid any misconception it is well to emphasize that the difficulties and the considerable fuel losses during exhausting experienced

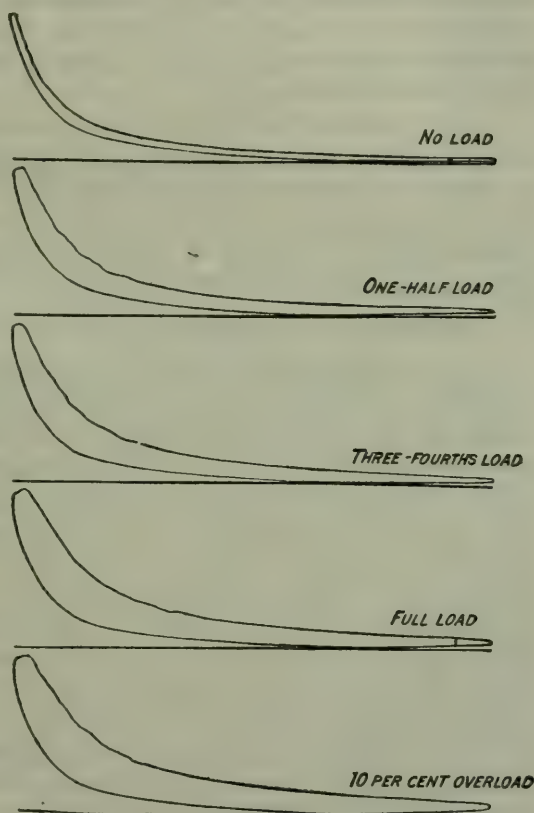


FIGURE 12.—Indicator diagrams of a Diesel engine having a four-stroke cycle.

with the usual two-stroke gas engine, chiefly on account of the successive scavenging and loading of the engine cylinder, are absent in the two-stroke Diesel engine. In this engine the same air pump does the scavenging and the charging, whereas in the two-stroke gas engine separate pumps are used for scavenging and for charging the cylinder with the gas-and-air mixture. These pumps consume considerable power, and it is difficult to prevent a part of the fresh gas-air charge from being expelled with the products of combustion during exhausting. Having to deal with inert air (air not mixed with any combustible gas) and a liquid fuel, the working process of the two-stroke-cycle Diesel engine is much simpler than that of the gas engine having a two-stroke cycle.

Engines with a four-stroke cycle, in sizes from 50 to 800 horsepower,<sup>a</sup> have been developed to a high degree of perfection, and are so simple in operation that little justification for the adoption of two-stroke engines in these sizes exists at present. The field of two-stroke types lies chiefly in engines of very small and very large powers, the latter from 1,000 horsepower upward.

#### ESSENTIAL QUALITIES OF ENGINES OF SMALL POWER.

Engines of small power should combine lightness with the greatest simplicity in design, to assure low manufacturing costs and ease of operation in inexperienced hands. In low-powered engines fuel economy and refinements in design are sacrificed to low first cost of engine. Nor is fuel economy of so much moment in small engines, as the total fuel consumption is a relatively small source of expense. Explosion engines having a two-stroke cycle best meet these requirements. Engines of this type have merely self-acting disk valves for admitting air to the crank case of the engine, and a fuel pump delivering a measured amount of fuel to the fuel-injection nozzle. This construction insures simplicity, but also results in low volumetric and mechanical efficiency and low fuel economy. Diesel engines, to insure high fuel economy, demand mechanically operated valves, and a high-pressure air compressor, which, together with the high compression and resulting high temperatures used, demand the highest grade materials and workmanship, involving a high cost per horsepower.

Although in European countries Diesel engines are manufactured in sizes as small as 15 horsepower there is little demand for this size in America, where fuel prices generally range lower, and where Diesel engines of 75 horsepower indicate the probable commercial limit in size. For large powers, from 1,000 horsepower upward, engines having a two-stroke cycle are preferred, as in these sizes limitations are put on the size of cylinder practical for engines having a four-stroke cycle by the high temperatures and pressures produced in such engines.

<sup>a</sup> The British horsepower is used throughout this publication. The British horsepower is 1.0139 metric horsepower, or 76.041 m. kg., whereas the metric is 75 m. kg. per second.

An increased use of material to withstand the greater total pressure in large cylinders merely accentuates the difficulty of cooling the cylinder, the cylinder head, and the piston, and of carrying off the heat fast enough through thick cylinder and cylinder-head walls. Stresses due to unequal expansion and contraction may easily lead to ruptures of vital engine parts, such as cylinder heads, pistons, and cylinders. Successful building of large Diesel engines must therefore be fortified by a great amount of practical experience, particularly in the rational design and selection of casting mixtures with correct chemical and physical properties.

Twenty-four inches in cylinder diameter represents the probable upper limit with air-cooled pistons, and 30 inches with water-cooled pistons, for Diesel engines having a four-stroke cycle.

#### COMPARATIVE ECONOMIES OF DIESEL AND OF EXPLOSION OIL ENGINES.

The use of explosion oil engines should be dictated entirely by their over-all economy. Although they are materially cheaper in first cost, they consume considerably more fuel and lubricating oil than Diesel engines, and their fuel consumption at fractional loads increases at a greater rate than does that of Diesel engines.

The difference in economy between explosion oil engines and Diesel engines is due not so much to a difference in thermodynamic cycles as in constructional differences, which decidedly favor the Diesel engines.

The higher the initial temperature and the lower the terminal temperature the more perfect will be the heat utilization. High initial temperature is a function of pressure. If the compression were carried as high in the explosion engine as in the Diesel, the explosion oil engine theoretically would show a slightly higher thermodynamic efficiency than the Diesel engine. The final pressure, when the fuel is burned at constant volume, would, however, be greatly increased above the compression pressure, or to about 60 atmospheres, with an accompanying rise in temperature far beyond that practicable with the materials of construction available. These limitations impose a lower compression pressure on explosion oil engines than on Diesel engines, so as to keep the maximum pressure and temperature within safe limits of permissible engine construction.

Thus with a compression pressure of 150 to 250 pounds per square inch the explosion pressure becomes 270 to 500 pounds per square inch; and with the instant ignition and burning of the previously vaporized oil common to explosion oil engines, a temperature of 2,300° to 3,150° F. is reached.

The Diesel engine has a higher but more gradually increasing compression pressure (450 to 500 pounds per square inch) which does not subject the engine to sudden shocks, with a resulting increase in temperature from atmospheric to about 1,000° F. The fuel is gradual-



ly injected as the piston moves from its top center (inner dead center) downward, the pressure remaining practically constant during the time of fuel admission. With a decrease in load, the time of fuel admission is also shortened, that is, the fuel supply is shut off sooner. Therefore the increase in temperature due to the burning of the fuel at constant pressure does not exceed that reached in an explosion engine, notwithstanding the higher compression pressure used in the Diesel engine, and the higher initial temperature caused by this compression. Thus, the temperature in a Diesel engine seldom exceeds 2,600° F., and reaches 3,000° F. only when the engine is overloaded.

If, then, the working cycles of the two types of engines are compared on the basis of compression pressures used, the Diesel engine is found to have a greater thermal efficiency, because it can work successfully with the higher compression pressure. This superiority is confirmed by comparative entropy diagrams.

Whereas in explosion oil engines the efficiency is influenced by the compression ratio alone, in the Diesel engine the efficiency is influenced by the compression and the cut-off ratios, the efficiency increasing with a decrease in the length of the cut-off or constant-pressure line. Thus, at fractional loads, the indicated thermal efficiency of Diesel engines increases, which partly offsets the loss in mechanical efficiency, that is, the increased fuel consumption for performing the internal work of the engine. This accounts for the very "flat" fuel-consumption curve of Diesel engines, which maintain an almost constant fuel economy over a fairly wide range in load. Thus at three-fourths load the increase in fuel consumption per brake horsepower-hour is only 2 to 5 per cent, and at one-half load 10 to 15 per cent greater than at full load in high-grade engines, which is in marked contrast with fuel increases in other prime movers. The ability to create higher initial temperatures, aside from increased thermal efficiency, enables the Diesel engine to burn a greater variety of fuels. In addition to the fuel being thoroughly atomized by highly compressed air, the heated oxygen has an augmented power of combining with the carbon and hydrogen in the fuel, so that the velocity of the chemical reaction at the high temperature in a Diesel engine is greatly increased. As a result the range of fuels suitable for the Diesel engine comprises such heavy liquid fuels as petroleum residues, coal-tar oils, and coal tars.

A serious fuel loss in explosion engines is frequently caused by the decomposition of the fuel oil sprayed into the hot ball. Various hydrocarbons are formed with a separation of carbon and oil soot; part of this coats the hot ball and the cylinder, and a larger part is expelled with the gaseous products of combustion. From time to time accumulated soot in the exhaust piping catches fire and burns, necessitating precautions against fire from this source.

Comparative economies of Diesel and of explosion oil engines are shown in Table 1 following. On a basis of oil costing \$1 a barrel,

TABLE 1.—*Comparative economies of Diesel and of explosion oil engines.*

[Cost of fuel oil taken as at \$1 a barrel of 320 pounds; cost of lubricating oil taken as at 35 cents a gallon.]

| Item.                                               | Diesel engine, European make. <sup>a</sup>                        |                                                                   |                                                                    |                                                                  |                                                                          |                                                                   | Diesel engine, American make. <sup>a</sup>                        |                                                                   |                                                                      |                                                                  |                                                                          |                                                                  | Explosion oil engine, European make. <sup>b</sup>                 |                                                                    |                                                                      |                                                                          |                                                                  |                                                                   | Explosion oil engine, American make. <sup>b</sup>                    |                                                                      |                                                                          |  |  |  |
|-----------------------------------------------------|-------------------------------------------------------------------|-------------------------------------------------------------------|--------------------------------------------------------------------|------------------------------------------------------------------|--------------------------------------------------------------------------|-------------------------------------------------------------------|-------------------------------------------------------------------|-------------------------------------------------------------------|----------------------------------------------------------------------|------------------------------------------------------------------|--------------------------------------------------------------------------|------------------------------------------------------------------|-------------------------------------------------------------------|--------------------------------------------------------------------|----------------------------------------------------------------------|--------------------------------------------------------------------------|------------------------------------------------------------------|-------------------------------------------------------------------|----------------------------------------------------------------------|----------------------------------------------------------------------|--------------------------------------------------------------------------|--|--|--|
|                                                     | Load.                                                             | Fuel per horsepower-hour.                                         | Per cent.                                                          | Barrels per year.                                                | Fuel cost per horsepower-year (\$8.760)                                  | Fuel per horsepower-hour.                                         | Load.                                                             | Fuel per horsepower-hour.                                         | Per cent.                                                            | Barrels per year.                                                | Fuel cost per horsepower-year (\$8.760)                                  | Load.                                                            | Fuel per horsepower-hour.                                         | Per cent.                                                          | Barrels per year.                                                    | Fuel cost per horsepower-year (\$8.760)                                  | Load.                                                            | Fuel per horsepower-hour.                                         | Per cent.                                                            | Barrels per year.                                                    | Fuel cost per horsepower-year (\$8.760)                                  |  |  |  |
| Fuel consumption or cost at different loads.....    | $\left\{ \begin{array}{l} 4/4 \\ 3/4 \\ 2/4 \end{array} \right.$  | $\left\{ \begin{array}{l} 0.42 \\ .44 \\ .50 \end{array} \right.$ | $\left\{ \begin{array}{l} 100 \\ 104.8 \\ 119 \end{array} \right.$ | $\left\{ \begin{array}{l} 863 \\ 678 \\ 685 \end{array} \right.$ | $\left\{ \begin{array}{l} \$11.50 \\ 12.05 \\ 13.70 \end{array} \right.$ | $\left\{ \begin{array}{l} 0.48 \\ .50 \\ .58 \end{array} \right.$ | $\left\{ \begin{array}{l} 4/4 \\ 3/4 \\ 2/4 \end{array} \right.$  | $\left\{ \begin{array}{l} 0.48 \\ .50 \\ .58 \end{array} \right.$ | $\left\{ \begin{array}{l} 100 \\ 104.2 \\ 120.8 \end{array} \right.$ | $\left\{ \begin{array}{l} 986 \\ 770 \\ 610 \end{array} \right.$ | $\left\{ \begin{array}{l} \$13.15 \\ 13.68 \\ 16.27 \end{array} \right.$ | $\left\{ \begin{array}{l} 4/4 \\ 3/4 \\ 2/4 \end{array} \right.$ | $\left\{ \begin{array}{l} 0.60 \\ .65 \\ .75 \end{array} \right.$ | $\left\{ \begin{array}{l} 100 \\ 108.3 \\ 125 \end{array} \right.$ | $\left\{ \begin{array}{l} 1,232 \\ 1,013 \\ 770 \end{array} \right.$ | $\left\{ \begin{array}{l} \$16.43 \\ 18.11 \\ 20.53 \end{array} \right.$ | $\left\{ \begin{array}{l} 4/4 \\ 3/4 \\ 2/4 \end{array} \right.$ | $\left\{ \begin{array}{l} 0.75 \\ .83 \\ .95 \end{array} \right.$ | $\left\{ \begin{array}{l} 100 \\ 110.7 \\ 126.7 \end{array} \right.$ | $\left\{ \begin{array}{l} 1,540 \\ 1,280 \\ 975 \end{array} \right.$ | $\left\{ \begin{array}{l} \$20.53 \\ 22.76 \\ 26.00 \end{array} \right.$ |  |  |  |
| Oil consumed for lubrication, gallons per year..... | 360 to 550.....                                                   |                                                                   |                                                                    |                                                                  |                                                                          |                                                                   | 360 to 550.....                                                   |                                                                   |                                                                      |                                                                  |                                                                          |                                                                  | 1,100 to 1,460.....                                               |                                                                    |                                                                      |                                                                          |                                                                  |                                                                   | 1,100 to 1,460.....                                                  |                                                                      |                                                                          |  |  |  |
| Cost of lubrication.....                            | \$128 to \$193 per year, or \$1.70 to \$2.60 per horsepower-year. |                                                                   |                                                                    |                                                                  |                                                                          |                                                                   | \$128 to \$193 per year, or \$1.70 to \$2.60 per horsepower-year. |                                                                   |                                                                      |                                                                  |                                                                          |                                                                  | \$380 to \$510 per year, or \$5.10 to \$6.80 per horsepower-year. |                                                                    |                                                                      |                                                                          |                                                                  |                                                                   | \$380 to \$510 per year, or \$5.10 to \$6.80 per horsepower-year.    |                                                                      |                                                                          |  |  |  |
| Cost of installation, including building.....       | \$90 per horsepower.....                                          |                                                                   |                                                                    |                                                                  |                                                                          |                                                                   | \$90 per horsepower.....                                          |                                                                   |                                                                      |                                                                  |                                                                          |                                                                  | \$60 per horsepower.....                                          |                                                                    |                                                                      |                                                                          |                                                                  |                                                                   | \$60 per horsepower.....                                             |                                                                      |                                                                          |  |  |  |

<sup>a</sup> 75-horsepower, one-cylinder, single-acting engine having a four-stroke cycle.<sup>b</sup> 75-horsepower, two-cylinder, single-acting engine having a two-stroke cycle.

the minimum yearly fuel cost in dollars corresponds to the number of barrels of oil consumed yearly. Oil usually costs more than \$1 a barrel, especially gas oils, which are the only oils suitable for use in explosion oil engines. On the other hand, residues and heavy fuel oils can be burned in the Diesel engine. Lubricating expense is also materially higher in explosion than in Diesel engines. With a fairly good load factor it will not take long to make up the difference in first cost by greater economy, particularly when the fuel cost is increased by long hauls.

### DIESEL'S USE OF CARNOT CYCLE.

Diesel's original ideal heat engine was planned to use the Carnot cycle. Aside from the difficulty of obtaining isothermal changes with gaseous fuels of low specific heat, the original Diesel cycle called for entirely impractical pressures and temperatures. Functional and structural difficulties of his engine as originally planned necessitated many mechanical and thermodynamical changes before the first successful engine was developed, in 1895. The first commercial engine, of 60 horsepower, was built early in 1898, and is reported to be in successful operation still. The same year, three engines were exhibited at the Munich exposition, one built by the Augsburg Machine Works, one by the Deutz (Otto) gas engine works, and the third by Krupp. Sulzer Bros. of Winterthur, Switzerland, built their first Diesel engine, of 20 horsepower, in 1896-97. The commercial manufacture and distribution of Diesel engines dates from that time, and at present engines representing more than two million horsepower are in operation.

The Carnot cycle may be briefly described by reference to figure 13, in which  $T$  represents absolute temperature and  $S$  represents entropy. The line  $da$  represents isothermal compression during which the heat of compression is to be absorbed as it is added; from  $a$  to  $b$  adiabatic compression sets in, during which no heat is to be abstracted or added. During the isothermal expansion, from  $b$  to  $c$ , heat is added to maintain constant temperature, and from  $c$  to  $d$  the expansion is to take place adiabatically in such a manner that the relation of the various compressions and expansions cause the final volume, pressure, and temperature to coincide with those at the beginning of the cycle.

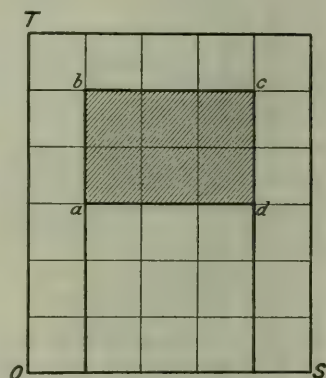


FIGURE 13.—Diagram explaining Carnot cycle.



This cycle demands that during the first part of the compression, heat must be perfectly carried away, that is, the working fluid and the surrounding material must be perfect conductors; and during the second period of the compression no heat must be carried away or added, thus calling for perfect nonconductors; similar opposing conditions are demanded during the expansion periods. Therefore it is not surprising that the theoretical requirement of a perfectly conducting and nonconducting material during different periods of the same cycle has never found practical fulfillment.

Diesel endeavored to meet the first condition by spraying water into the cylinder, the spray to be stopped during the adiabatic compression. During the isothermal expansion the fuel was to be injected into the cylinder to produce the required heat. The adiabatic compression had therefore to be high enough to heat the air to a point at which it would ignite the fuel. In the first Diesel engine planned, coal dust was to be the fuel. The adiabatic terminal pressure was to reach 250 atmospheres, and was to drop to 90 atmospheres with the termination of the isothermal combustion of the fuel. The cylinder was not to be water-cooled, but to be insulated against all radiation of heat. The thermal efficiency was to be 73 per cent. On account of the high pressure and temperatures involved and the small amount of useful work obtained, the original cycle was changed to that which has been in use for the past 20 years. It still remains the nearest approach to the Carnot cycle. The Diesel engine with its 45 per cent indicated thermal efficiency is the most efficient heat engine of to-day. The efficiency of the Diesel cycle is about 60 per cent, of which 75 per cent is realized in indicated work (45 per cent net indicated thermal efficiency) and 60 per cent in effective mechanical work (36 per cent net effective thermal efficiency) in high-grade, four-stroke engines at full load. Two-stroke engines have a slightly lower efficiency.

## DETAILS OF DIESEL ENGINES.

Diesel engines are built in both vertical and horizontal units, each type showing some differences in construction in the frame, the valve gear, the fuel injector and the lubricating arrangements.

### VERTICAL ENGINES.

Vertical engines are built as one cylinder and as multicylinder units. The structural details of the bedplate of such engines vary according to whether it is to support the closed crank case or individual cylinder frames, so-called A frames. For supporting the closed crank case the entire top joint of the bedplate is planed to insure an oil-tight joint between it and the crank case; for supporting individual cylinder frames the bed is provided with carefully machined

pads or surfaces to which the A frames are bolted. The bed is cast integral with the crank pit and the seats to receive the main-shaft bearings. The pit is as a rule used as a reservoir for collecting the spilled and surplus lubricating oil. Either method of building the engine has its advantages, which are more those of manufacture than of any inherent superiority of one arrangement over the other. A bedplate for supporting a closed crank case is shown in Plate I, A.

#### ADVANTAGES OF INDIVIDUAL CYLINDER FRAMES.

The choice of individual cylinder frames enables the manufacturer to build with greater ease two, three, four, or even five and six cylinder units from a smaller number of standardized cylinders and valve gears. With an engine having a closed crank case this is not so readily and cheaply accomplished. Inasmuch as most engines are used for the generation of electric current with direct-connected generators, four-cylinder units are necessary to obtain the desired cyclic regularity for satisfactory generator performance in engines having a four-stroke cycle, especially when two or more units are to operate in parallel. Multicylinder units also permit a higher number of revolutions, by shortening the stroke without necessarily increasing the piston speed of these units. Such features combine to produce a compact, relatively light engine. The closed crank case is the construction favored for medium and high speed engines, with which forced-feed lubrication is used. There is no spilling and spattering of lubricating oil, and the engine and engine-room floor are readily kept clean. The forced-feed lubricating system, at first used only on high-speed units, has proved so satisfactory and so economical in the use of lubricants that it is used increasingly on medium-speed and low-speed engines. In Europe manufacturers have adopted this closed crank case construction for units up to 1,000 horsepower.

In larger engines, which require the use of A frames, the panels are connected with steel sheets provided with large doors between the individual frames, so as to obtain the advantages of a closed crank case and of forced-feed lubrication.

On the bedplate is also mounted the multistage air compressor, usually in a vertical position to preserve the symmetry of the engine.

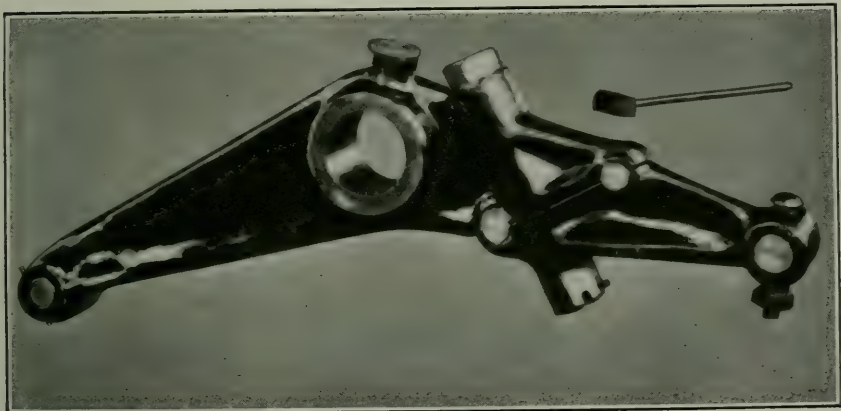
#### CONSTRUCTION OF CYLINDERS AND CYLINDER FRAMES.

The cylinder frame receives the cylinder liner which has at the head end a heavy flange fitted to the cylinder frame. A registered joint is either placed wholly in the top of the flange of the cylinder liner or is divided over the top of the cylinder frame and the liner. Some constructors prefer the latter method, to insure a water-tight



A. BED PLATE OF 120-BRAKE-HORSEPOWER VERTICAL ENGINE WITH CLOSED CRANK CASE.

Housings for main bearings are being scraped to true seat for bearing shells by turning in the housings a cast-iron shell, ground to exact diameter, covered with "blue grease" used in scraping off high spots. Some of the shells and cups are shown.



B. CONSTRUCTION OF EXHAUST-VALVE LEVER PERMITTING EASY REMOVAL OF VALVE.





and a gas-tight joint. The cylinder head is provided with a corresponding ring that fits into the registered joint, soft-copper packing rings being inserted between the surfaces.

The top end of the cylinder frame must be so designed as to provide ample metal, as in it are secured the stud bolts that hold the cylinder head and are subject to heavy stress.

The cylinder liner is provided in the middle with a machined rib or ring to center the liner and also to give it lateral support against a similar ring cast integral with the cylinder frame. The bottom part of the liner has another ring provided either with a gland ring or a groove into which fits a circular rubber packing ring. Both methods are used to insure a water-tight joint between mantle and liner. The cylinder liner is fixed only at the head end, and is therefore free to expand in the other direction.

The use of separate castings for the mantle and for the cylinder liner is important, as the cylinder liner becomes much hotter than the mantle, so that the unequal stresses produced in the two are likely to fracture a cylinder if cast integral with the mantle. Separate liners can be designed and made of the most suitable material; in case of excessive wear or fracture, they can be cheaply and easily replaced. On account of the rapid alternations of exceedingly high and low temperatures, cylinder liners are preferably made of a special close-grained cast iron of high tensile strength—capable of withstanding a load of 40,000 pounds per square inch—and able to withstand severe shock tests.

For cylinder lubrication the cylinder frame and the liner are perforated in four places, and oil passages consisting of small copper pipes are inserted between the mantle and the liner, and bridge the water space. These oil passages are in the same plane in the middle part of the cylinder, being spaced  $90^{\circ}$  apart, and  $45^{\circ}$  removed from the axis of the main shaft. As the front and the back cylinder walls receive the principal pressure, the oil feeds in these cylinders are sometimes set closer together, so as to feed the oil more freely to the surfaces most taxed.

The length of the cylinder liner is determined by the length of the stroke and of the piston. The length of the piston is governed according to whether it is a trunk piston or whether it is guided by a crosshead. The piston may extend one-fifth of its length out of the cylinder. Cylinder frames and liners of engines having a two-stroke cycle differ from those of engines having a four-stroke cycle, previously described, in that the bottom part has exhaust ports and water-cooled passages for conducting the products of combustion into the exhaust pipes. In some engines the scavenging air ports are also placed in that part of the cylinder.

### HORIZONTAL ENGINES.

The frames of horizontal Diesel engines do not differ materially from those of ordinary horizontal gas engines. They are built either as one or two cylinder frames. If four-cylinder units are desired, two-cylinder frames are united by placing the flywheel and the driving pulley or electric generator between the two frames. The cylinder mantle and the crank pit, with bearing supports, are cast integrally.

The frame is provided with machined pads, to which the air compressor is bolted. The compressor is usually driven from an overhanging crank pin at the end of the main shaft. According to whether the connecting rod is connected to a crosshead carried in a guide or whether the trunk piston, with its piston pin, serves as a modified crosshead guided by the cylinder, the cylinder frame is proportionately long or short. To collect the spent cylinder oil, a cast-iron rib extends vertically upward and across the frame, separating the crank pit from the forward part of the cylinder frame and preventing the bearing oil collected in the crank pit from becoming mixed with the used cylinder oil.

The cylinder liner is constructed as for vertical engines and is set into the frame in much the same manner. Cylinder frames for engines having a two-stroke cycle are built along similar lines as for vertical engines, or the cylinder and jacket mantle, with cored exhaust ports and water-cooled conduit for exhaust gases, are cast in one piece, with the cylinder liner extended so as to be slipped into and bolted to a supporting ring which is part of the main engine frame; the overhanging cylinder frame is then supported by a suitable footing or column. The engine frame is designed to receive the injection air compressor as well as the scavenging air pump.

### CONSTRUCTION OF CRANK SHAFT.

The high pressures common to Diesel engines demand proportionately heavy crankshafts. These are forged from the solid, a ductile low-carbon steel being used. Some manufacturers use nickel-steel shafts. For large engines, or for an engine made of two half units, the shaft is composed of more than one piece, although the common practice for stationary engines is to use one-piece shafts, even if four cranks are used.

### METHODS OF LUBRICATION.

When the shaft and the connecting rods are not drilled for forced-feed lubrication, it is usual to provide each crank with a centrifugal oiling ring to deliver the oil to the crank pins. For this purpose the crank pins are drilled, the drilled passages receiving the oil from the rings and delivering it to the pin brasses. Oil-drip rings turned out of the solid material of the shaft confine the oil to the main bearings and lead it to the oil reservoirs in the crank pit. On the shaft is



mounted the helical gear for engaging a like gear on the governor shaft and driving it, the gears dipping into the oil. A crank pin for driving the air compressor is part of the main shaft and is lubricated in the same manner as the other pins. The flywheel and the driving pulley are usually mounted on an enlarged part of the shaft, the extension of which is carried in an outboard bearing.

The main bearings for this type of shaft are either provided with an oiling ring or are constantly flooded with oil under pressure issuing from a pipe in the cover of each bearing. The surplus oil is collected in the crank pit and is filtered and cooled, and is returned to the bearing by a pressure oil pump. A circulating system of this kind insures positive lubrication of the main bearings, whereas oiling rings may possibly stick and through failure of lubrication cause the burning of a bearing.

Another system of lubrication, essential to high-speed and even medium-speed engines, and, on account of its satisfactory operation and sparing use of lubricants, favored increasingly for high-grade low-speed engines, is illustrated in figure 14. Drilled passages run from the middle of the main bearings diagonally through the middle of a journal and the webs of the cranks and issue in the middle of the crank pins. The connecting rod is hollow. Whenever these passages register, a part of the oil, aside from lubricating the main and the crank-pin bearings, is forced through the hollow connecting rod and lubricates the piston-pin bearing. The oil is initially forced under pressure through the covers of the main bearings. In other engines the oil is supplied through the bottom of each main bearing, permitting a simple arrangement of the oil piping and obviating the necessity of breaking pipe connections when the bearing covers are removed. The surplus oil is collected in the crank pit, flows to a filter, through a cooler, and is returned to the bearings by a positive-pressure pump, driven direct from the main engine either by gearing or side cranks. Figure 15 illustrates such an oiling system.

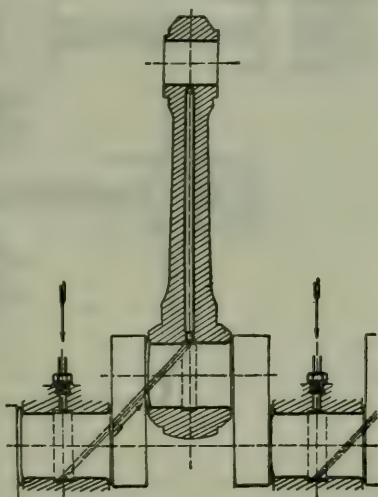


FIGURE 14.—System of lubrication especially adapted to high-speed engines.

The lubricating system last described effectively lubricates the piston-pin bearing, which is of particular importance, as its position inside the trunk piston and in close proximity to the piston head



with a wiper is fastened to the inner wall of the trunk piston. With every stroke of the piston a measured supply of oil furnished by a separate pump plunger is carried to the piston-pin bearing.

A method of driving a rotary lubricating oil pump on a vertical engine is shown in Plate II, *A*.

### MAIN BEARINGS.

The main bearings are usually semicylindrical, babbitted, cast-steel shells, carried by carefully machined seats cast integral with the bedplate, the shells being held down by their covers. They have no wedge adjustment, but rest rigidly in their seats. In practice it has been found difficult to keep numerous adjustable bearings in perfect alignment, and inequalities in adjustment of the different bearings have frequently led to broken shafts. With the bearings once carefully machined and aligned in the shop the wear over all bearings is uniform and light in high-grade engines, in which the bearings are flooded with oil from a forced-feed system, uniform bearing pressure being insured through an equally proportioned supply of fuel oil furnished to each cylinder. Thin brass shims are placed between the bearing shells; to take up wear a shim is removed from each side of each bearing. By raising the shaft slightly the shells can be rolled out for rebabbitting. The shop method of insuring alignment of the bearing seats is shown in Plate I, *A* (p. 20). The seats are chalked, and any inequality is apparent on the carefully machined arbor. Any roughness is then removed by scraping.

### CONNECTING ROD.

The connecting rod is forged of soft, low-carbon steel and has a hollow center in engines using forced-feed lubrication. Hollow rods are often used, especially in small engines, to decrease weight. The crank-pin end is usually provided with a marine head, the boxes of which are either lined with babbitt or bronze, the head proper being of cast steel, and is bolted to the connecting rod. Brass shims are usually placed between the head and the rod to permit adjustment and insure the desired compression. The piston-pin end of the rod is usually slotted out of the solid to receive an adjustable box made of cast steel or cast iron lined with babbitt and bronze. Adjustment is made either by shims held by adjusting screws or by wedges. The positive rigidity obtained with square shims and adjusting screws is usually given preference over the wedge adjustment. For large engines a marine head for the piston end is sometimes preferred as offering greater ease of adjustment; however, it offers no advantages over the closed head when it has to be removed. The piston and rod of either type are pulled through the top of the cylinder



and the rod is removed by knocking out the piston pin. In horizontal engines the pistons are pulled out of the open bottom of the cylinder, so that the heads do not have to be removed, an advantageous arrangement. Some types of vertical engines are so built as to permit pulling the piston from the bottom of the cylinder.

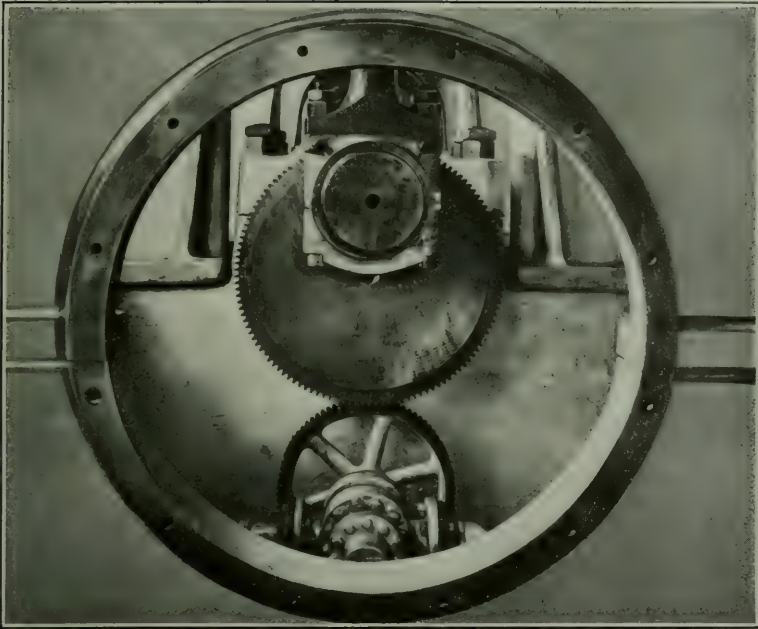
### PISTONS.

On account of the relatively short connecting rod and the high pressure common to Diesel engines, the resultant side pressure on the cylinder wall is high. Therefore, the trunk pistons must be correspondingly long to keep the unit pressure within safe limits and to reduce excessive cylinder and piston wear. For this reason crossheads and guides were used during the earlier development of Diesel engines in Europe. On account of its many advantages the trunk piston has been adopted and is used in engines, even of 250 horsepower per cylinder and in units of 1,000 horsepower. The trunk piston is more easily provided with a greater bearing surface than a crosshead, thus insuring less wear; the lubrication under pressure of a cylindrical guiding surface is more effective than with open guides. The piston moves over perfectly cooled walls, whereas the crosshead tends to heat more readily and when once hot is not easily cooled; moreover, water-cooled crossheads complicate construction. The use of crossheads and guides greatly augments the height or the length of the engine, increasing its cost. For large engines this construction is used, as it affords greater accessibility and ease of adjustment, the crosshead guide in such engines being water cooled.

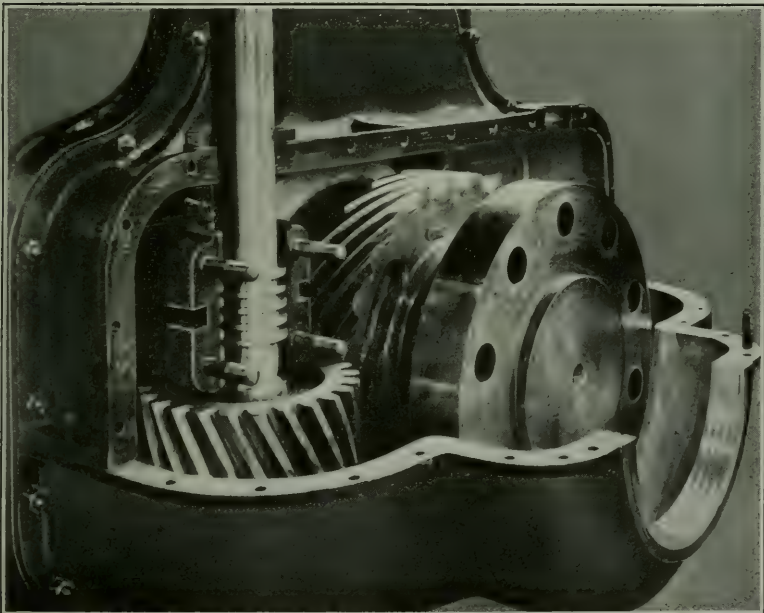
To confine the wear to an easily and cheaply replaceable part of the piston, in some constructions the piston on the side subjected to the reaction pressure is provided with soft, cast-iron wearing shoes, which can be raised slightly by the interposition of thin sheet-metal shims, when shoes and cylinder are worn. However, this construction is not extensively used, as, with good design and workmanship and thorough lubrication, the wear is slight and is noticeable only after years of work. Cylinder wear is usually due to an excessive sand or ash content in the fuel oil used and not to piston side pressure.

The cylinder liner is cast sufficiently heavy to permit of reboring.

From five to seven snap rings are used to seal the cylinder. To hold the rings closed they are pinned together at the lap joints. They are pinned to the piston by a small dowel set in a hole in the grooves. Such fastening is always necessary in two-cycle engines to prevent the ends of the snap rings from shifting around the piston and thus traveling across the exhaust ports and catching in them while they are being uncovered or covered by the piston.



A. METHOD OF DRIVING ROTARY LUBRICATING OIL PUMP IN A VERTICAL ENGINE. PUMP IS SHOWN AT BOTTOM OF LARGE CIRCULAR OPENING.



B. VERTICAL GOVERNOR SHAFT ON A VERTICAL ENGINE, DRIVEN THROUGH HELICAL GEARS FROM THE MAIN SHAFT.





At the bottom of the piston another ring is placed, serving as an oil-wiping ring and preventing the drawing of splashed oil from the crank case and the reciprocating parts into the engine cylinder. This feature is of importance in a high-speed engine with a closed crank case.

Pistons of small size are cast in one piece; those of larger engines are cast in two pieces. The limiting size for the one-piece and the two-piece construction differs widely with different builders. In the two-piece construction the piston top with the snap rings comprises about the upper one-third of the piston. In the lower piece is placed the gudgeon pin. That part of the piston subjected to intense heat and great wear by the impinging of the fuel and air jets can thus be cheaply replaced. The top of the piston is either a plane surface or has a concave depression. To reduce the clearance space to a minimum and to obtain the required compression, the piston is permitted to come close to the head when at its top center by cutting out recesses in the top of the piston for the air inlet and exhaust valves. To prevent excessive wear of the central top part of the piston, upon which the fuel and the injection air impinge, a nickel or steel plate is sometimes cast into that part. Other constructions provide a small cone to distribute the fuel radially over the entire combustion space and to improve the combustion of the fuel. The construction of the piston is largely a matter of practical experience. The interior of the trunk is provided with reinforcing ribs for strength as well as for radiating heat readily. On account of the repeated heat changes and high pressures, the piston is heavily taxed. Notwithstanding, all material must be used judiciously and in the right places, as any excess iron will serve only as a heat accumulator and subject the piston to severe internal stresses caused by differences in temperature in different parts of the piston. A light piston is also desirable to reduce the weight of the reciprocating masses.

Large pistons are jacketed and cooled by oil, water, or air and water. The oil or water is either circulated through the jacket space, or the water is sprayed with air against the inside surface of the piston top.

In some engines oil is used for piston cooling, to prevent contamination with water of the lubricating oil in the crank case, should the moving pipe connections supplying the cooling medium spring a leak. However, the use of oil as a cooling medium is not general; as its specific heat is only about half that of water. Water is also far superior for transmitting heat. As satisfactory mechanical constructions have been developed that insure a continuous supply of cooling water to the piston with small likelihood of leaks, the use of oil has been superseded by that of water even by builders who have long adhered to the former practice.

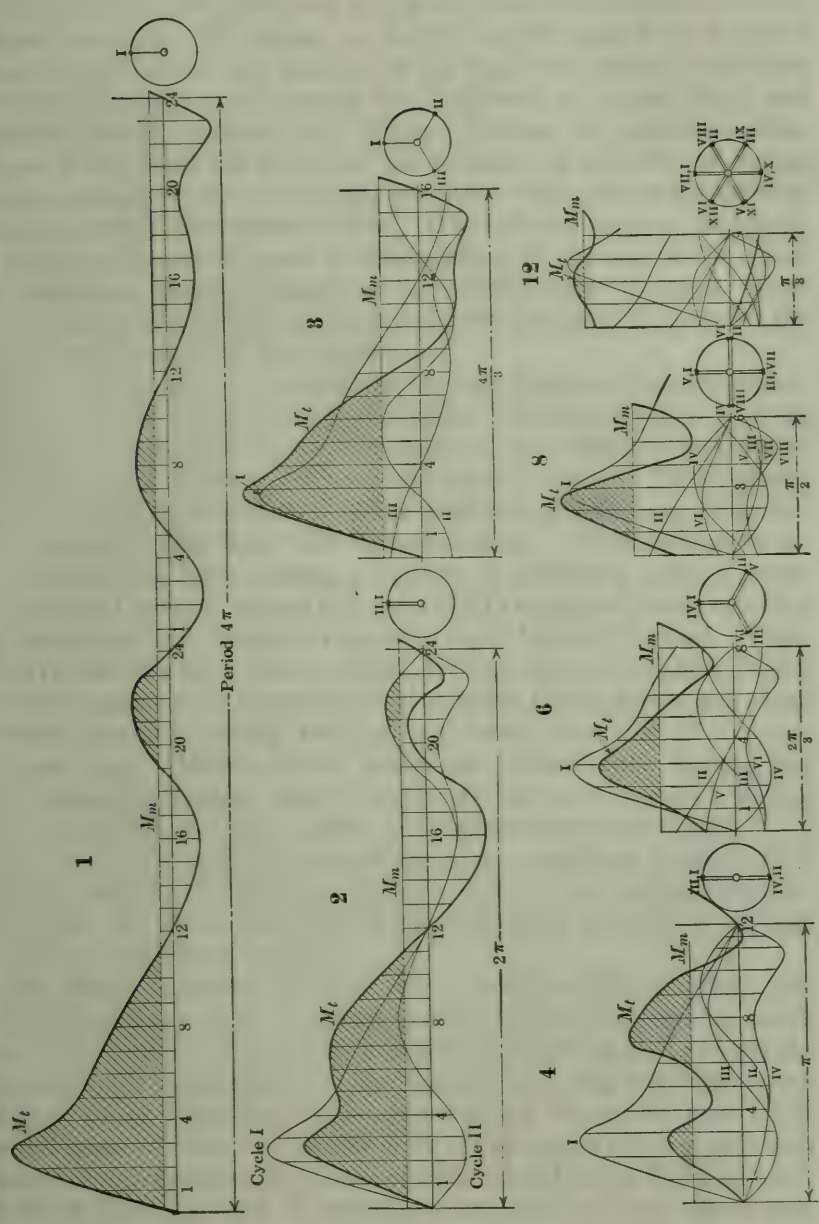
The water for piston cooling may be circulated through telescopic pipes, with the stuffing box a moving part of the piston, or the stuffing box may be attached to the frame and the water be supplied to the piston through hollow walking arms through which the water flows to the pipes leading into the piston. A pump may be actuated from the crosshead and the water be carried to the piston through the hollow piston rod. In another cooling system water is sprayed by air against the heated piston surface, not enough water being used to fill the water space of the piston. The excess water drains off through a pipe surrounding the spray pipe, no stuffing box being used.

There is no uniform practice among manufacturers regarding the size of engine at which a change from air-cooled to water-cooled pistons is made. Some use water-cooled pistons when the cylinder reaches 125 horsepower. Others build air-cooled pistons for cylinders of 250 horsepower. In construction involving a closed crank case, which gives little opportunity for air-cooling of the piston, the former practice is favored, whereas for horizontal engines in which the interior of the trunk and the part of the piston extending outside of the cylinder are exposed to the air the higher limit may be used.

Pistons of two-stroke cycle engines, in which the scavenging air is admitted through ports in the bottom of the cylinder, have special shaped tops to direct the flow of the scavenging air.

In horizontal engines, the ratio between the length of connecting rod and that of the stroke is generally greater than in vertical engines; a ratio of 3 is common, so that the reaction pressure against the side of the cylinder is usually less than in vertical engines, even allowing for the additional pressure due to the weight of the piston, which is always slight compared with the total side pressure on the cylinder.

The cylinders are lubricated by providing each cylinder with a force-feed pump operated by a reducing motion from the end of the piston, or with a multiple-plunger pump driven by an eccentric device or direct from the cam shaft, one plunger being provided for each working cylinder and one for each air-compressor cylinder. In some engines two plungers per cylinder are provided, one for each pair of the four oil feeds grouped around the cylinder. Vertical two-stroke Diesel engines usually have a separate oil feed above and below the exhaust ports to prevent any excess of lubricating oil from being swept through the exhaust ports. Horizontal two-stroke engines usually have the exhaust ports in the sides of the cylinder, the parts of the cylinder serving as top and bottom guides being solid; the lubricating oil can then be so distributed over the bearing surfaces as to occasion little if any loss through the exhaust ports and separate oil feeds are not needed.



CRANK-EFFORT DIAGRAMS FOR FOUR-STROKE ENGINES WITH 1, 2, 3, 4, 6, 8, AND 12 CYLINDERS.





## FLYWHEEL.

Flywheel construction does not differ from that common to gas-engine practice. To insure a fairly uniform speed, a flywheel becomes necessary. On account of the high compression and mean effective pressures, the cranks are subjected to severe duty and transmit correspondingly high efforts. During the power impulse, excess power must be stored in the flywheel to be released during the compression stroke and during the other two unproductive strokes in engines with a four-stroke cycle. One-cylinder engines, especially if great uniformity of speed is demanded, as in driving two or more alternators in parallel, require an excessively heavy flywheel. Concentration of all the power in one cylinder also greatly increases the comparative weight of the engine.

Commercial as well as technical reasons, therefore, impose limitations on the size of single-cylinder units. These seldom exceed 150 horsepower and then are usually confined to horizontal engines for industrial uses allowing a liberal variation in speed. Two-cylinder units are built in sizes of 50 to 300 horsepower, and three and four cylinder units in sizes from 100 horsepower up. Multicylinder units, by dividing the work among a number of cylinders, have a better rhythm and greatly reduce the individual tangential forces received by the crank, and therefore require a much lighter flywheel. Thus, the flywheel of a one-cylinder engine, if used on a four-cylinder engine with the same size of cylinder and four times larger power, will cause a cyclic regularity nearly 400 per cent better than that of the one-cylinder unit. Multicylinder units are considerably lighter; the smaller-sized cylinders allow shorter strokes and therefore higher rotative speeds, and these reduce the required flywheel weight.

Plates III, IV, and V show how the cyclic regularity is affected by (a) the number of cylinders; (b) the cycle of the engine, whether two-stroke or four-stroke and whether single or double acting; and (c) the setting of the cranks.

The abscissas denote the time cycle during which the forces act, and the ordinates show the forces acting tangentially on the crank pin. The number of cylinders and the setting of each of the cranks are shown. The area between the base line and  $M$  is the mean crank effort or power furnished by the engine during the entire cycle. The shaded areas represent the excess work of the engine developed during periods of maximum work, which must be stored in the flywheel to be given off during periods of negative work; that is, when the engine produces insufficient power and consumes power.

In Plate III, the numbers 1, 2, 3, 4, 6, 8, and 12 denote the number of cylinders of same size in the four-stroke engines compared. The shaded areas become successively smaller with an increase in the

number of cylinders, notwithstanding the fact that the mean crank effort corresponding to the number of cylinders is proportionately greater. If all these engines were to have the same degree of regularity, the flywheels would have to be dimensioned proportionately to the shaded areas, clearly illustrating the much reduced flywheel weight required by multicylinder units. As previously mentioned, such units give much greater uniformity of motion and rhythm with a corresponding decrease in the weight of reciprocating masses and dimensions of units, thus permitting great compactness with relatively small head room. These are the factors principally responsible for the general adoption of multicylinder units and not the inability to concentrate only small power in one cylinder, as European manufacturers have developed cylinders that yield 2,000 horsepower.

The crank-effort diagrams in Plate IV are of single-acting two-stroke engines with 1, 2, 3, 4, 6, and 8 cylinders. These engines require still lighter flywheels, mainly because they receive a power impulse per cylinder during each revolution, instead of every other revolution as in engines having a four-stroke cycle, represented in Plate III.

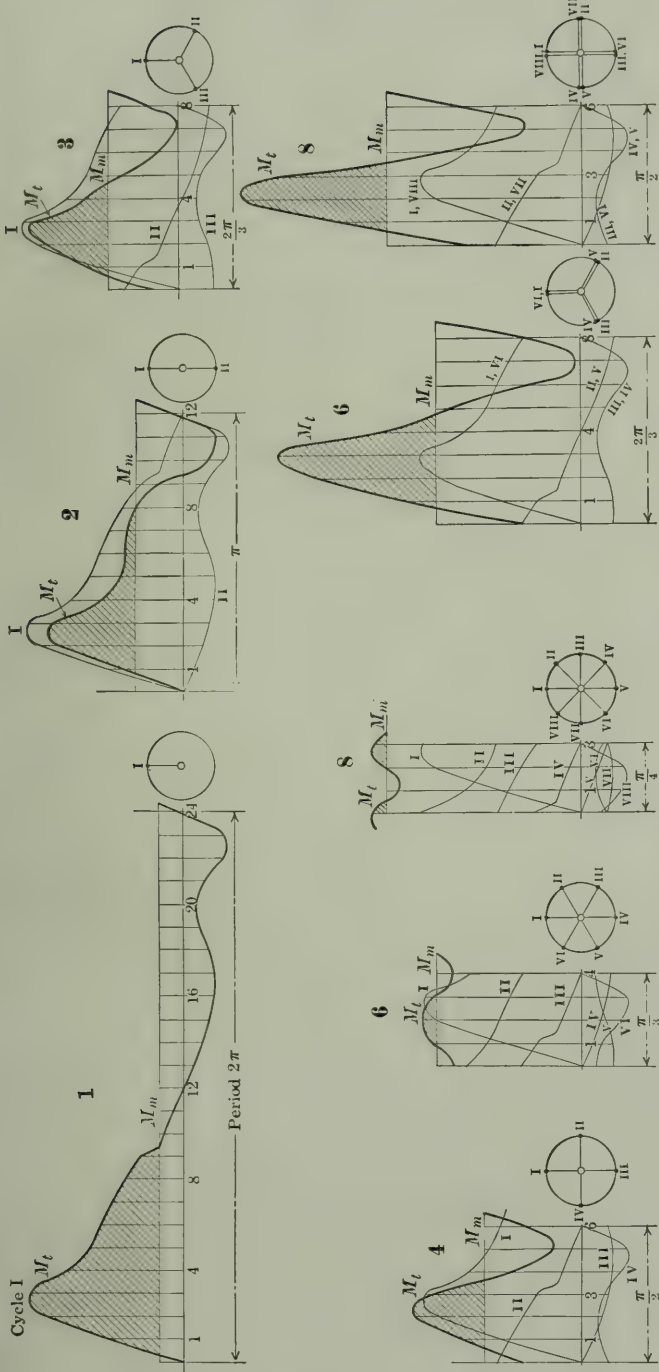
Although in all the diagrams of Plate III and in the first six of those in Plate IV the cranks are shown set so that the power impulses are equal time intervals apart, the last two diagrams in Plate IV representing six and eight cylinder engines illustrate a crank position in which two power impulses are transmitted to the shaft simultaneously. This setting increases correspondingly the excess work to be stored in the flywheel and requires a proportionately heavier flywheel.

In Plate V are shown diagrams of double-acting, two-stroke engines with 1, 2, 3, 4, and 6 cylinders; two six-cylinder engines are shown, one receiving only one working impulse at a time per cycle, the other receiving two simultaneous working impulses per cycle. These diagrams show that double-acting, two-stroke engines require the smallest flywheel weights.

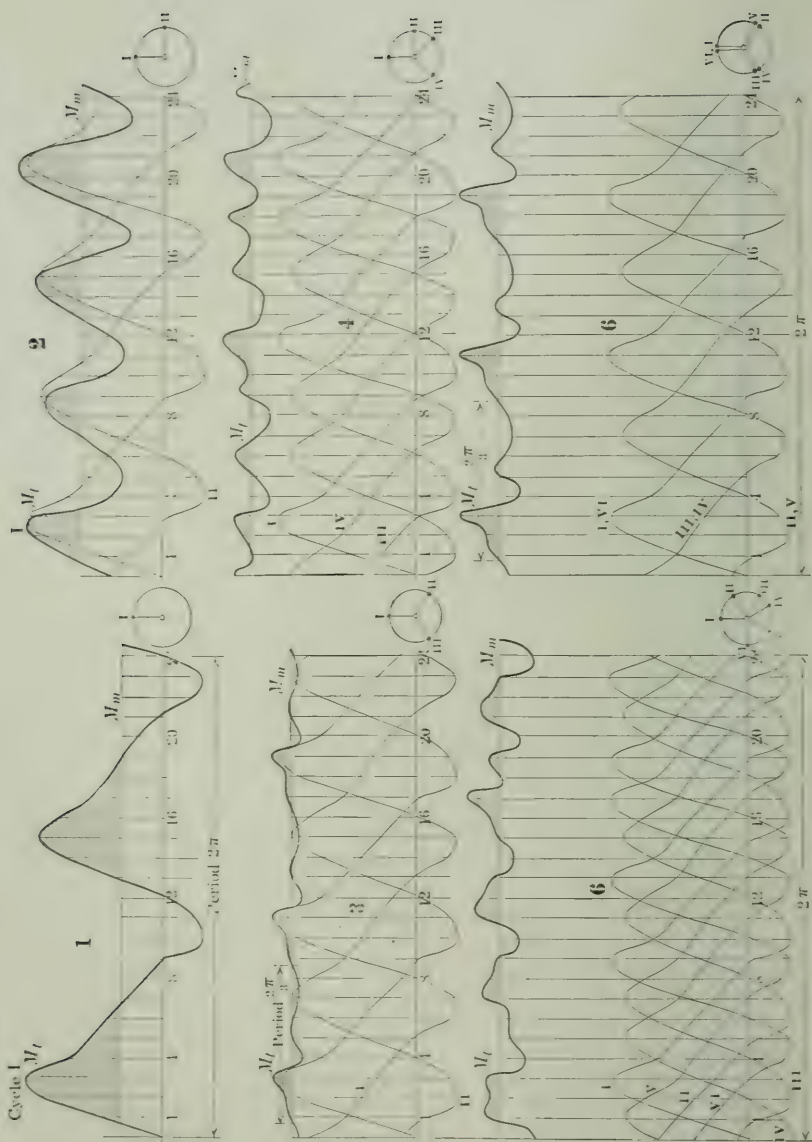
A one-cylinder, single-acting, two-stroke engine generally requires about the same flywheel effect as a two-cylinder four-stroke engine; and a given number of cylinders in a double-acting four-stroke engine imparts the same regularity to it as does an equal number of cylinders in a single-acting two-stroke engine.

The irregularity factor may be defined as the ratio of the difference between the maximum and the minimum velocities to the average velocity of the flywheel during one revolution and is expressed by the formula  $\frac{V_{\text{maximum}} - V_{\text{minimum}}}{V_{\text{mean}}}$ ; it therefore expresses the maximum variation in velocity during one revolution and is also known as the pulsation of the prime mover.





CRANK-EFFORT DIAGRAMS FOR SINGLE-ACTING TWO-STROKE ENGINES WITH 1, 2, 3, 4, 6, AND 8 CYLINDERS.



CRANK-EFFORT DIAGRAMS FOR DOUBLE ACTING TWO STROKE ENGINES WITH 1, 2, 3, 4, AND 6 CYLINDERS.

The maximum variation from uniform speed differs widely for different classes of work required of the engine. The irregularity factor may vary as follows:

For driving pumps, punches, shears, and machines requiring a low degree of regularity, one-twentieth to one-thirtieth.

Machine tools, looms, and paper machinery, one thirty-fifth to one-fortieth.

Spinning machinery, according to class of work, one-sixtieth to one one-hundredth.

Direct-current generators, one one-hundred-and-fiftieth.

Alternating-current generators, one two-hundred-and-fiftieth to one three-hundredth.

Making allowance for the different classes of power service that Diesel engines have to perform, manufacturers usually provide standard light, medium, and heavy flywheels for all engines, the medium and heavy flywheels of smaller engines being also available as light and medium flywheels for progressively larger machines. The set of flywheels used on one-cylinder engines is also used on multicylinder units with cylinders of same size.

One-cylinder units should not be used if the permissible variation has to be less than one-fiftieth; two-cylinder units may be used if the factor of irregularity does not have to be less than one one-hundredth; one two-hundredth is the limit for three-cylinder units, and one three-hundredth or less for four-cylinder units. Naturally, three and four cylinder units can perform the work of one or two cylinder units with a much lighter flywheel. The foregoing figures apply also to engines having a two-stroke cycle but with half the number of cylinders.

Limitations as to weight and dimensions of engine and flywheel, floor-space requirements, the more perfect balancing of the much-reduced reciprocating masses, and the great compactness will often decidedly favor the multicylinder unit, especially when the Diesel engine is to be directly connected to an alternating-current generator and two or more units are to operate in parallel. Multicylinder units also permit a high rotative speed, which lowers the flywheel weight and increases the permissible variation of the engine through the reduction of the number of poles in the generator.

#### **CYLINDER HEAD.**

In engines having a four-stroke cycle the usual practice is to place all valves in the cylinder head; in engines having a two-stroke cycle the exhaust valves, and in some engines the scavenging valves also, are placed in the cylinder and the other valves in the head.

The cylinder head of vertical engines (fig. 16) is a cylindrical water-jacketed body, with flat bottom and top, and is secured to the cylinder frame with liberally proportioned stud bolts. Openings for the different valve cages in the top of the head are grouped symmetrically whenever feasible; the fuel-injection valve is usually in the center,



and the air and exhaust valves on each side, in the same cross-sectional plane and in line with the axis of the engine. The opening for the starting valve is usually placed in front of the central opening, in the same plane, and at right angles to the plane through the other three valve openings. When the available head area is too small to permit the fuel-injection, air, and exhaust valves to be placed in the same plane, the fuel valve is placed slightly forward, permitting the other valves to be placed closer together, so as to allow the required area.

A flat, smooth, bottom surface in a head results in a simple combustion chamber not split up with ribs and pockets. Such a surface is a great aid in reducing the clearance space to the required minimum and affects favorably the combustion of the fuel. Such heads can

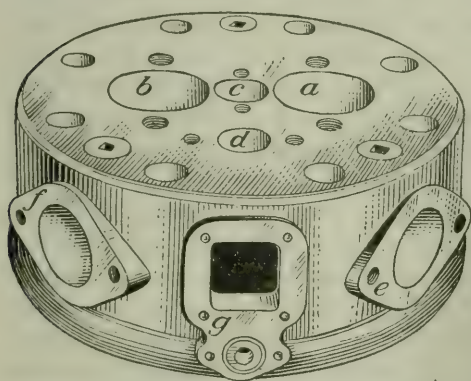


FIGURE 16.—Cylinder head of vertical Diesel engine. *a*, opening for cage for air-admission valve; *b*, opening for cage for exhaust valve; *c*, opening for fuel valve; *d*, opening for starting valve; *e*, facing for air-pipe connection; *f*, facing for exhaust-pipe connection; *g*, facing for water-pipe connection and handhole.

be easily designed for vertical engines, permitting the use of vertical valve stems, which insure positive and easy seating.

The advantages of a flat head are considered sufficient by some manufacturers of horizontal engines to outweigh the advantages of vertically seating valves. Others attach more importance to vertical valves and follow gas-engine practice in the construction and operation of the valves and the cylinder head. The air valve is placed in the top of

the head and the exhaust valve in the bottom, the fuel valve being in front, in the center.

The starting valve is then placed either in the front of the head or on the side halfway between the air and the exhaust valves. The use of a gas-engine head provides a larger water space; however, this is much constricted by the numerous walls surrounding the valve openings if the valves are all placed in the top or end of the head. Efficient cooling of the head is then rendered more difficult. Hence, it is advisable, if a supply of pure water is not available, to use distilled water for cooling the cylinder head and the cylinder, and to recool the water for reuse; otherwise trouble may be experienced from precipitation of lime and magnesia salts allowing the head to become overheated and causing it to crack.

The placing, form, and direction given to the air and the gas passages differ in accordance with the space available and the design adopted for the air-intake and the gas-exhaust pipes.

The heads of two-stroke engines have generally the same cylindrical form as those of four-stroke engines, but in the former there are no openings for the exhaust valves, which are placed in the lower part of the cylinder.

The scavenging valves are placed in the head, and as the volume of air required for a given cylinder volume is 1.2 to 1.4 times greater

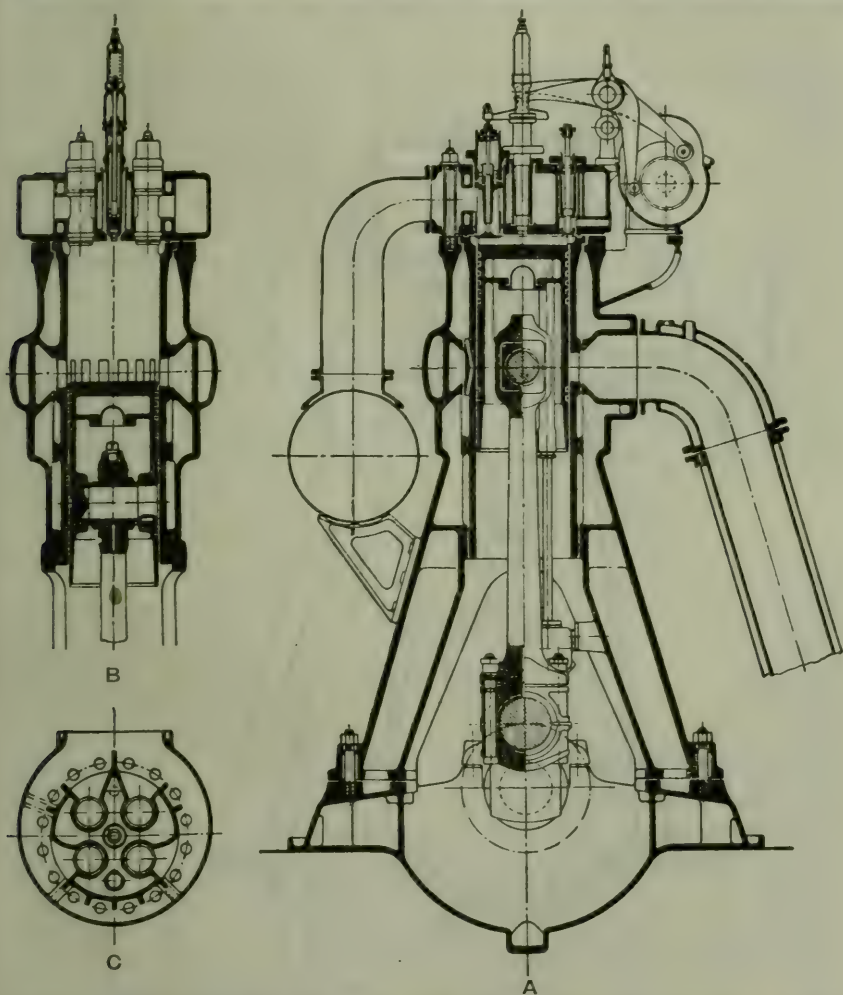


FIGURE 17.—Two-cycle engine with air admission through cylinder head, showing scavenging air valves. Exhaust ports uncovered by piston are shown at A.

than for engines having a four-stroke cycle, and as the time available for the scavenging and the refilling of the cylinder with fresh air is short, large valves are needed. The area of the head sets definite limits for the size and placing of these valves. To facilitate their simultaneous opening and simplify the valve gearing, the valves are grouped in pairs, and four valves are symmetrically

distributed over the top of the head (fig. 17). The fuel and the starting valves are placed as in engines having a four-stroke cycle. The placing of so many valves in the cylinder head greatly restricts the water space and renders cooling more difficult; the valve area is also limited by the head area, and high air velocities and correspondingly high scavenging air pressures are necessary. These

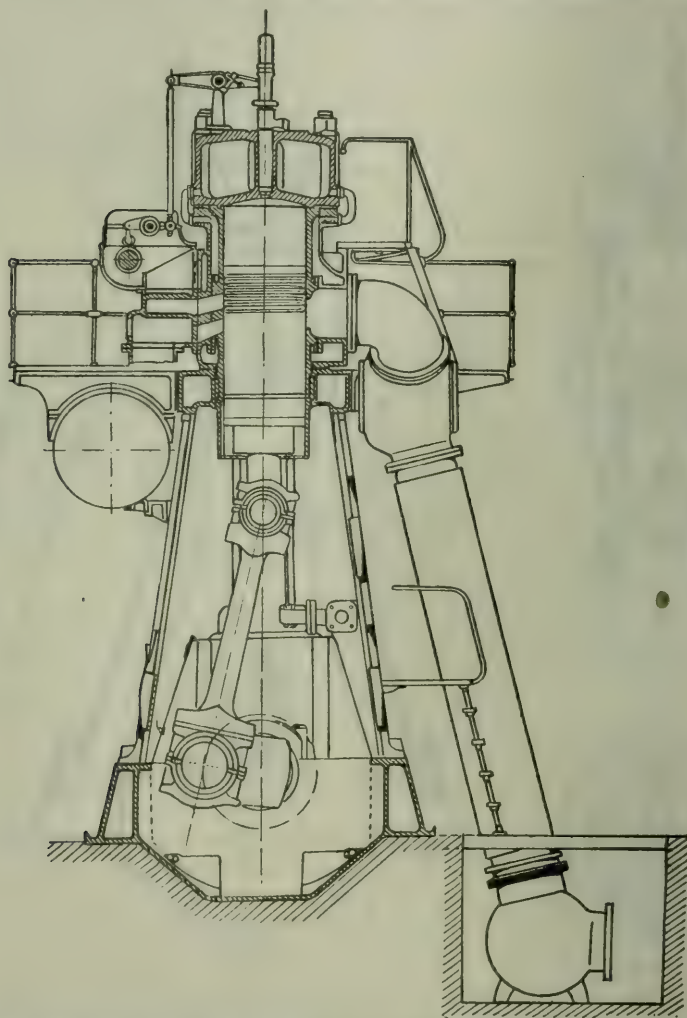


FIGURE 18.—Two-stroke engine with two tiers of air valves. Air valves opposite exhaust ports.

disadvantages are partly offset by the effective scavenging obtained with valves opening into the top of the cylinder. A more recent engine construction simplifies the cylinder head by placing the exhaust ports as well as the ports for the scavenging air in the middle of the cylinder (fig. 18). Two tiers of scavenging ports are provided, of which the lower one is uncovered by the piston, whereas



the upper is controlled by a valve actuated by the cam shaft. The lower tier of ports is opposite the exhaust ports, and in direct communication with the air receiver when uncovered by the piston. After the piston has traveled past the exhaust ports and the lower row of scavenging ports and closed these, the upper row of air ports is still uncovered, and air can be forced into the cylinder, increasing the specific output of the engine. The valves controlling the upper tier of ports are in a region of low pressure, and therefore low temperature. During the periods of compression and high temperature these valves are covered by the piston, whereas with the scavenging valves in the head there is a certain amount of danger of leaking valves letting fuel vapors or gases enter the scavenging air receiver and cause explosions. The valves closing the upper tier of air ports prevent the entrance of exhaust gases into the scavenging air receiver before the pressure of the exhaust gases is reduced to atmospheric pressure with the uncovering of the exhaust ports by the piston. The absence of scavenging valves in the cylinder head simplifies its form greatly, as all surfaces can be equally and thoroughly cooled; and with the elimination of internal ribs between the valve seats, stresses due to unequal expansion are avoided. Only the fuel-injection, starting, and relief (safety) valves are in the cylinder head.

#### AIR INLET AND EXHAUST VALVES.

Air inlet and exhaust valves (fig. 19) are usually housed in flanged cages bolted to the cylinder head. A cage contains the valve seat and the valve-stem guide. The upper part of the barrel is machined and serves to guide the carefully centered spring cap. The latter is screwed to the valve stem and holds the spring in place. The valves are generally made of cast iron, the stem being of steel. It is usual practice to provide valves, except those of the smallest sizes, with removable valve seats. Air and exhaust valves seldom differ, except that for engines with more than 30 or 35 horsepower per cylinder the valve-stem guide is water-cooled. Four-stroke engines are seldom built in sizes requiring the water-cooling of the exhaust valves, and two-stroke engines exhaust through ports uncovered by the piston. Figure 20 shows the customary valve construction. The spring cap has a concave depression, into which the jointed ball of the actuating lever fits.

In engines fitted with gas-engine heads, the usual practice is to house only the air valve (placed in the top of the head) in a cage. After the air valve with cage has been removed, the exhaust-valve stem, disk, and seat are accessible through the opening in the air-valve cage and can be removed through it. In two-cycle engines with gas-engine head, only the scavenging valves are in the head, and

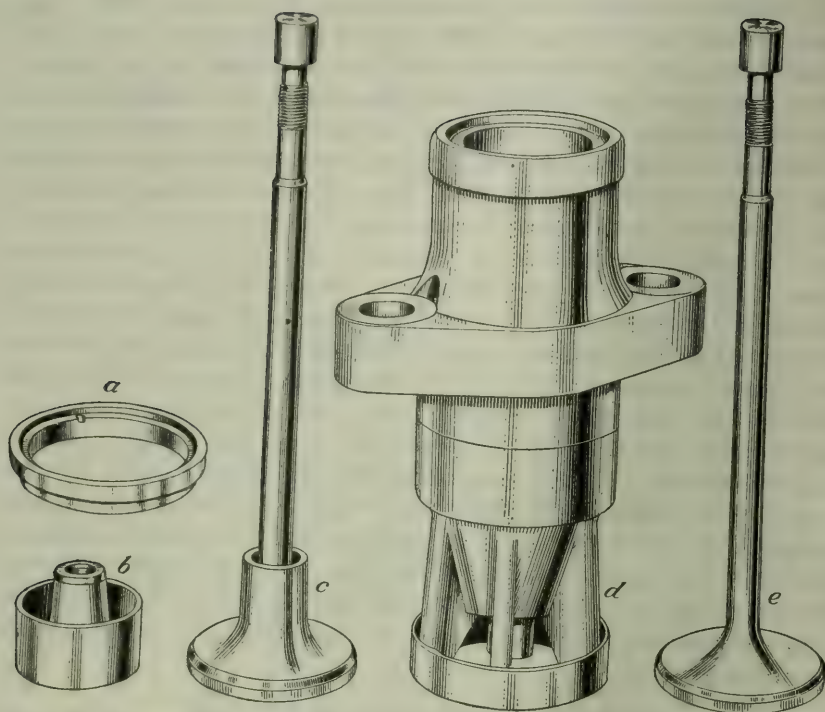


FIGURE 19.—Valves and valve fittings. *a*, removable valve seat; *b*, spring cap; *c*, exhaust valve in stem; *d*, valve cage; *e*, air admission valve in stem.

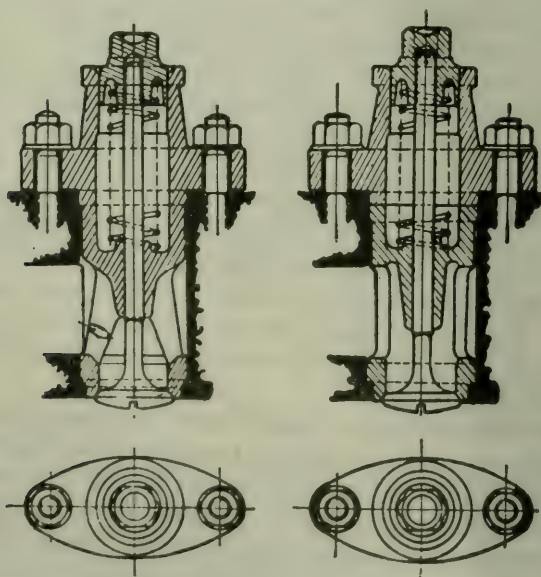


FIGURE 20.—Customary valve construction of Diesel engines.

are on its top and bottom, one directly above the other. In this type the valves are similar and are housed in cages. As only fresh air is admitted through them, and they are not fouled by the products of combustion, they seldom need to be removed. To retain a perfect seat, each valve disk is occasionally ground in its seat. The disk is provided with two holes for the insertion of a special key, or the valve stem is slotted so that a bar can be inserted, for turning the valve disk back and forth; a small amount of emery dust and oil is used in grinding.

The starting valve admits air under high pressure (500 to 600 pounds per square inch) into the engine cylinder whenever the engine is started. Hence the valve disk must seat perfectly to prevent air leakage when the valve is closed. It is usually constructed as a balanced valve; a part of the valve stem is enlarged and provided with packing rings, the air admitted pressing equally on both sides of this piston. A strong spring insures the tight seating of the valve.

Some engines are equipped to start with air at a pressure of about 250 pounds per square inch. The starting valve is fitted with a plunger that closes the valve as soon as the cylinder pressure exceeds that of the air used for starting.

### FUEL VALVES.

#### TWO FUNCTIONS OF FUEL VALVES.

Next to the fuel pump, the fuel-injection valve is the most important and characteristic part of a Diesel engine, and on its correct construction greatly depends the satisfactory operation of the engine. It performs two distinct functions. First, it has to admit the liquid fuel into the engine cylinder at the proper time, and must therefore be an accurately timed valve. Second, it has to introduce the liquid fuel into the cylinder in a completely atomized form, and must therefore be an efficient atomizer.

The first requirement is fulfilled by storing around the tip of the fuel needle a drop of oil, which is forced ahead of the injection air into the heated atmosphere of the engine cylinder at regularly recurring intervals, and thus initiates the ignition. The second requirement is accomplished by dividing the fuel supply into numerous small fuel particles in the fuel valve, preceding atomization and injection.

#### ATOMIZATION OF FUEL.

As has already been explained, highly compressed air is used for atomizing the fuel oil; incidentally it serves the purpose of injecting the fuel oil into the cylinder during the process of atomization. In the atomizing nozzle of the fuel valve a part of the pressure of the



injected air is changed into velocity, which reaches its maximum at the orifice of the nozzle. Owing to the expansion of the air at the orifice there is a decided drop in temperature at this point, where the ignition of the fuel is to be initiated. This localized lowering of the temperature at the nozzle orifice is accentuated during periods of low load, when the fuel supply is diminished proportionately to the engine load, whereas the injection air supply remains constant; that is, the same as at full load, because the fuel-valve needle stays open during 10 to 15 per cent of the stroke of the engine, irrespective of the load. The relation of fuel supply to air supply is therefore changed with a change in load. It is consequently more difficult to initiate the ignition at partial than at normal engine loads. A correctly designed fuel valve must be so constructed that unfailing fuel ignition at the proper instant will take place for every working impulse. Moreover, the atomization and injection of the fuel must be gradual during the entire period the needle valve is open.

Preheating of the injection air to counteract a cooling effect during fuel injection not only constitutes an element of danger owing to possible fuel ignition in the fuel valve, but results in an uneven, knocking operation of the engine as a result of explosion-like ignition of the fuel.

#### COMMON TYPE OF ATOMIZER.

The elements of a common type of atomizer are shown in figure 21. It comprises a fuel needle *a*, guided for a great part of its length in a cylindrical barrel *b*. The needle and the guide are carried in a flanged cage, which is slipped into the central passage in the cylinder head and is bolted to the latter. The outer diameter of the needle guide is smaller than the internal diameter of the cylindrical valve cage. The annular space *c* thus formed around the needle guide is connected with the injection air receiver and is at all times filled with compressed air. The needle guide terminates and seats against a corrugated cone *d*, provided with about 20 slots, each one-sixteenth inch wide. The cone fits the cone-shaped end of the valve cage. Above the cone are a number of perforated plate rings, *e*, so placed that the perforations of each succeeding ring are staggered with regard to those in the ring above. Each ring is provided with 18 to 20 apertures five sixty-fourths to three thirty-seconds inch in diameter. An atomizer tip is screwed over the end of the valve.

The needle valve is held against its seat by a powerful spring, provided with a suitable spring cap, a bonnet, and exterior adjusting nuts. In the body of the valve an oil passage is drilled to deliver the fuel on top of the perforated rings. In other types, to avoid the cost of drilling these long, small passages, two concentric spaces are used—the inner for air, the outer for fuel—formed by inserting a cylinder over the needle guide, thus dividing the annular space around the guide

into two compartments. Holes in the bottom of this cylinder permit the fuel to enter the inner air space and to be distributed over the perforated plates. The measured quantity of fuel oil proportioned to the engine load is deposited by the fuel pump in the fuel-valve chamber before the valve needle is opened.

The fuel oil is broken into innumerable small globules and runs from plate to plate and down into the grooves of the cone, and is dis-

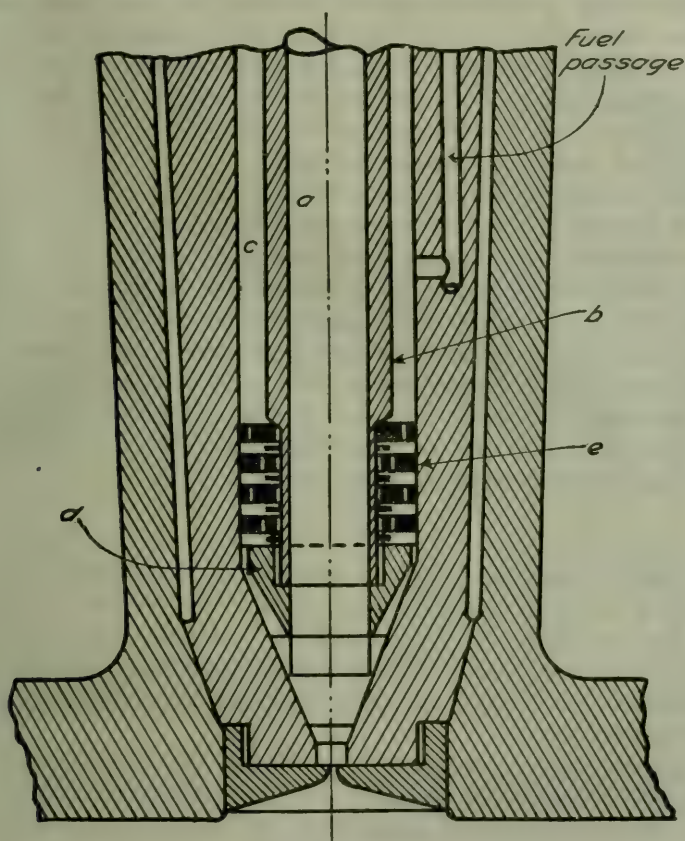


FIGURE 21.—Common type of atomizer. *a*, fuel needle; *b*, barrel; *c*, annular space; *d*, corrugated cone; *e*, perforated plate rings.

tributed over and covers the surfaces of the plates and grooves, and a small quantity collects around the root of the fuel-needle point. When the needle valve is opened, the injection air sweeps this part of the fuel before it, propagating the ignition. The air attains its greatest velocity when issuing through the nozzle. The sluggish, viscous oil particles, being in a quiescent state, resist this sudden acceleration with their inherent inertia and move relatively much more slowly than the air. The rapidly moving air currents tear these

particles to pieces, and when the air currents have attained their highest velocity at the atomizer orifice they nebulize the fuel. A small proportion of the fuel still remains behind and flows down to the top of the seat of the fuel-valve needle to initiate the ignition of the next fuel charge.

#### ADJUSTMENT OF PRESSURE OF INJECTION AIR.

Fuel valves of this type require a change of pressure of the injection air with a variation in load; that is, the pressure must lessen as the load of the engine is decreased if perfect combustion with colorless exhaust and smooth running of the engine is to be obtained. The pressure may vary from 40 to 70 atmospheres, depending on the engine load and the nature of the fuel.

If the pressure and quantity of injection air were not reduced during periods of low load when the fuel quantity is much less, the air would sweep through the perforated rings and the cone and clean them of all fuel, leaving none to accumulate around the fuel needle point. When the needle was opened the next time, the ignition might either fail completely or be delayed and cause a series of rapid, dull explosions. If the engine misses ignition, it will slow down, and the fuel pump will deliver an increased quantity of fuel oil, which will be ignited regardless of the high injection air pressure, but such speed variations and engine operation should be avoided.

#### ADJUSTMENT BY HAND.

A common way to vary the pressure of the injection air is by throttling down the air taken into the air compressor and adjusting it to the varying loads of the engine. This throttling must be done by the engine operator. The air pressure also has to be adjusted to the properties of the fuel oil, some oils requiring a higher injection air pressure than others.

In other engines provision is made for changing the height to which the valve needle is lifted. A lever is actuated to lift or to lower the fuel needle, and this lever is fulcrumed on a pin the journals of which are eccentric to the lever bearings, so that by rotating this pin to various positions, the length of opening of the fuel needle may be adjusted to the engine load. A small hand lever is keyed to one end of the pin and is provided with a "click" to hold the lever as set.

#### AUTOMATIC ADJUSTING DEVICES.

Such hand adjustments of the injection air pressure present no particular difficulty in small engines, but in large engines, especially those working with rapidly fluctuating loads, it is obviously impossible for the operator to adjust the air pressure quickly enough to meet the constantly changing requirements of the engine. The work of the air



compressor consumes 10 per cent and more of the power output of the engine. In large engines it is therefore desirable to proportion the volume of injection air to the engine load. This is accomplished by a patented design in which the engine governor influences not only the volume of the fuel supply, but also the volume of the injection air; the governor likewise affects the length of time during which the fuel valve is open, the period varying in proportion to the load on the engine. The governing devices are illustrated in figure 22. In the figure, *a* is a throttling valve actuated by the engine governor *b*, proportioning the amount of air taken into the scavenging pump *c* to the engine load, a part of the air from which is compressed further in the compressor *d*, which furnishes the injection air. The pressure and the quantity of the injection air are thus controlled by the engine governor. Similarly, the varying air pressure influences the travel of the piston in the servo-motor *e*, which, acting through auxiliary levers and rollers, varies their position relative to the cam. The travel of the lever actuating the needle valve is thereby varied and the length of time for which the needle valve is open is controlled.

A similar arrangement can be used in engines having a four-stroke cycle, the volume of air going to the injection air compressor being throttled by the governor, and the servo-motor being operated with air from the first stage of the compressor.

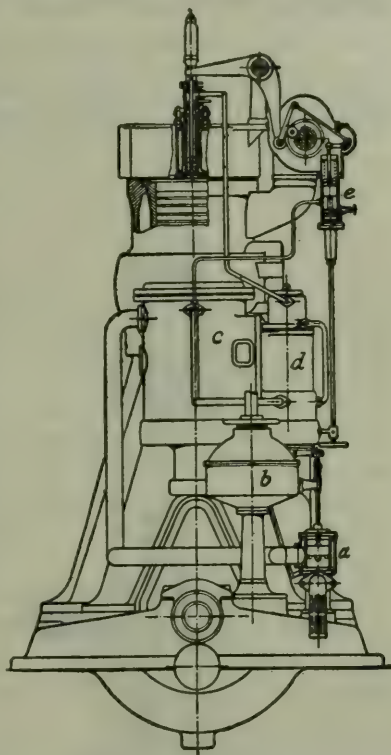


FIGURE 22.—Patented throttling devices for Diesel engines. *a*, throttling valve; *b*, engine governor; *c*, scavenging pump; *d*, compressor; *e*, servo-motor.

#### METHODS OF FUEL INJECTION.

##### HELSELMANN'S FUEL INJECTOR.

To insure an unfailing correctly timed ignition of the fuel charge, Hesselmann, of the Swedish Diesel Engine Co. (Aktiebolaget Diesel's Motorer), developed the fuel injector shown in figure 23. In this valve the fuel is admitted close to the end of the fuel needle. For this purpose the valve has two annular air spaces, connected with air passages at the top. The fuel is deposited in the bottom of the

outer space and under the influence of the full injection air pressure is forced through a number of small passages, from the orifices of which it is swept off by the passing air currents in the inner space, as soon as the valve needle is opened. The fuel is in this manner thoroughly nebulized, and a small quantity left behind accumulates around the root of the needle point to initiate the ignition of the succeeding fuel charge with the opening of the needle.

#### NEEDLE VALVES WITH ANNULAR FUEL PASSAGES.

In other forms of fuel valves, the same object is obtained by the use of annular fuel passages around the conical needle-valve seat, the

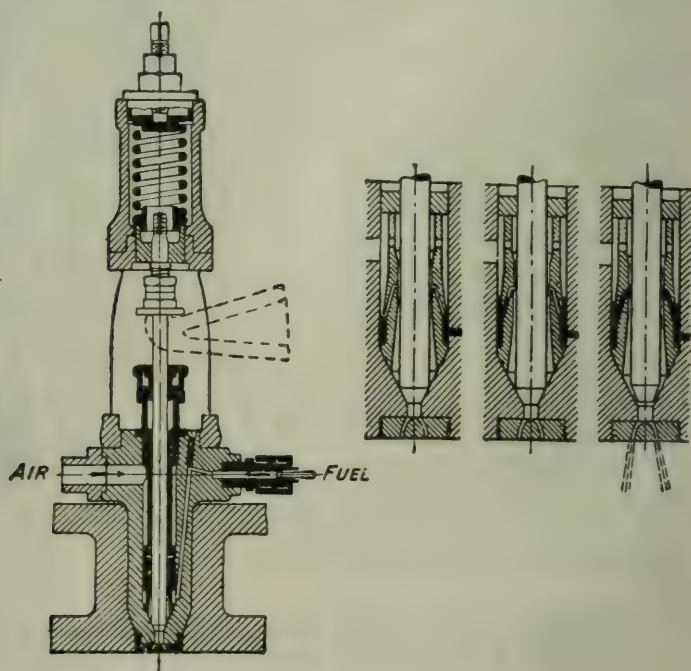


FIGURE 23.—Fuel injector developed by Hesselmann.

oil in which is under the influence of the injection air pressure. When the fuel valve is opened, part of the pressure is changed into velocity; the resulting differential in pressure between the injection air that presses on the fuel and that which travels through the nozzle forces the fuel out of the annular passages through radial openings into the air stream, by which it is picked up and atomized. A small quantity of oil left around the needle point on the closing of the valve is used for igniting the next charge.

Fuel valves of this type permit considerable load fluctuations without necessitating appreciable change in the injection air pressure. They do not, of course, effect a reduction in the quantity of air used for injecting the fuel, which would be practically constant at all loads

unless controlled by one of the methods previously described. In large engines, the saving of power effected by supplying only the required volume of injection air is important, although it necessitates adjustment to produce the most perfect ignition and combustion of the fuel. This consideration led to the automatic mechanical control of the injection air supply for large engines (fig. 22).

In the use of the fuel valves described, the fuel has to be pumped into the fuel chamber against the injection air pressure, which may vary from 40 to 70 atmospheres, the higher pressure corresponding to full load, so that the fuel pump must always work at a pressure appreciably higher than the injection air pressure. Such valves are also known as closed-nozzle valves, because the nozzle must be kept closed with the needle, except during the extremely short period of fuel atomization.

#### USE OF OPEN-NOZZLE FUEL VALVE.

Another type of fuel valve, known as the open-nozzle type (also called the Lietzenmayer system, after its designer), uses an open fuel nozzle, as its name indicates. In fuel valves of this construction the fuel does not work against the high injection air pressure, but is subject merely to the small pipe resistance and low lift necessary to deposit the measured amount of fuel in a recess of the fuel-valve chamber in front of the open nozzle; this is usually done during the suction stroke of the engine, when there is the least resistance (pressure) in the valve chamber. When the time for the injection of the fuel arrives, a valve, similar to the needle valve above described, is opened, admitting the injection air into the fuel chamber; the air, in passing over previously deposited fuel, impinges on it, picks it up, and, forcing it into the atomizer, nebulizes it. A valve of this type is shown in figure 24.

#### EQUIPMENT OR METHODS PERMITTING USE OF COAL TAR AND COAL-TAR OILS.

All of the valves mentioned work well with fuels that are easily ignited. Certain liquid fuels, such as coal tars and coal-tar oils, are ignited with difficulty, and when the engine is cold (before it is started) or at low load ignition fails entirely. The size of the engine cylinder, as well as the speed of the engine, have considerable influence on the success of the ignition. Small cylinders radiate the heat relatively faster than large cylinders, the ratio of the area of cylinder walls and head to the volume of the combustion space being greater than in large engines. Consequently the combustion space of small engines is cooled faster than that of large engines, especially during periods of low load. High engine speed shortens the time available for ignition and burning of the fuel, which in some engines must be done in less than one one-hundredth of a second.



## STARTING ENGINE WITH GAS OIL.

To permit the burning in Diesel engines of coal-tar oils (a by-product of coking and gas-works operations and a distillate of coal tar), two methods have been used. One is to start the engine with gas oil and after it has become hot change to tar oil. The governor

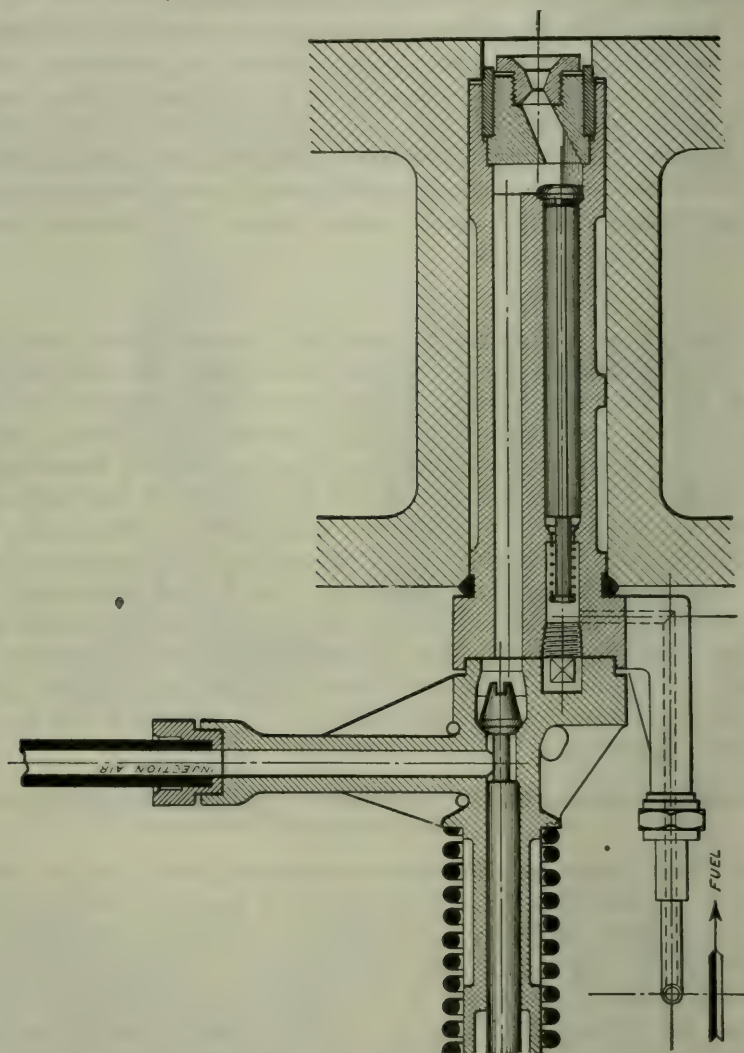


FIGURE 24.—Open-nozzle fuel valve.

and the fuel pump are so designed that with a lowering of the load the fuel is changed to gas oil and vice versa. This method has the disadvantage of causing an unnecessarily large quantity of gas oil to be consumed if the engine works a large part of the time at fractional loads.

## USE OF SEPARATE FUEL PUMPS.

In another system separate fuel pumps are used for the gas oil (also referred to as ignition oil, because it initiates the ignition) and the coal tar or tar oil. The two oils are stored separately in the fuel valve, the ignition oil being so placed around the edge of the end of the valve needle that it will always enter the combustion space first, and thus initiate the ignition. The fuel is injected with compressed air as in the other valves previously described. The proportion of gas oil to tar oil used varies with the load. At no load the gas oil amounts to as much as 20 per cent of the tar oil; at quarter load it drops to 15 per cent, at half load to less than 10 per cent, at three-quarters load to 4 per cent, and at full load and at overload to about 3 per cent. During average operating conditions the consumption of gas oil seldom exceeds 10 per cent, and is usually 7 per cent. The combined fuel consumption, on a basis of heating value, is nearly the same as if only gas oil were used. Both pumps are, of course, under the influence of the engine governor. The nicety and sensitiveness of this highly successful process may be realized from the fact that the volume of gas oil injected ahead of the tar oil is exceedingly small. Thus for a 50-horsepower cylinder the total volume of fuel oil injected for one power stroke at full load is only 2 c. c., or a cube of the size shown in figure 25. The part of the cube corresponding to the volume of gas oil used during usual load conditions (80 fuel charges producing 160 revolutions per minute) represents only 0.06 to 0.2 c. c.

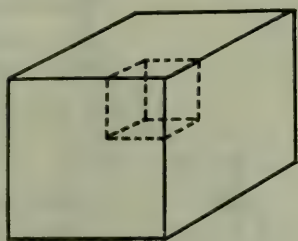


FIGURE 25.—Proportionate volumes of tar oil and of gas oil used in one power stroke of 50-horsepower cylinder at full load. Large cube represents volume of tar oil; small cube represents volume of ignition (gas) oil.

If the fuel oils, tar, or tar oils are highly viscous and do not flow readily to the fuel pump and to the fuel valve and resist atomization, gas oil is used to start the engine and run it until hot, and the hot discharge water from the engine jacket is used to heat the viscous oil to increase its fluidity. Then the gas oil is displaced by the viscous oil. This system requires a separate small tank for the gas oil, in addition to the one used for the viscous fuel oil, and the pipe system must be so arranged that either fuel can be supplied to the fuel pump. When the engine is to be shut down, the gas oil should be switched on and the engine operated on gas oil until all of the viscous oil has been displaced. This arrangement works satisfactorily except when the operator forgets to change over to gas oil before stopping the engine or in the event of an unforeseen sudden shutdown owing to some engine

trouble. These contingencies are so rare that separate fuel pumps and pipe systems for the two kinds of fuel are seldom provided. The method mentioned should be used on all fuel oils the viscosity of which at ordinary temperature ( $20^{\circ}\text{C.}$ ) exceeds  $3^{\circ}$  Engler.

#### RETARDING INJECTION AIR AND FUEL.

In the cylinders of large engines the fuel needle can be so governed that the entrance of the injection air and fuel into the combustion space is retarded and gradual, preventing excessive use of injection air cooling of the atmosphere at the point where the ignition must be maintained; by these means tar oil can be ignited without the use of ignition oil.

The methods of using tar oils described are of no immediate importance in the United States, where an abundant supply of petroleum

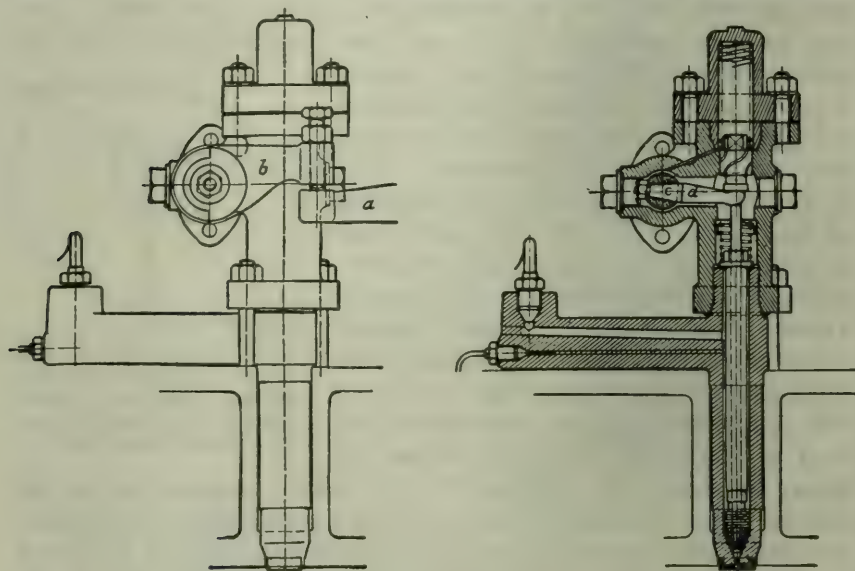


FIGURE 26.—Type of fuel-valve construction permitting easy removal of fuel needle or upper part of valve; *a*, rocking arm; *b*, auxiliary lever; *c*, horizontal pivot joint; *d*, lever which transmits lifting motion.

products (gas oils and residues) is cheaply obtainable, and coal tar commands relatively higher prices. In Europe, particularly Germany, petroleum residues and gas oils are expensive, and the abundant supply of tar oils from by-product coke ovens and coal-gas works justifies their use by the method outlined. The prospects for the Diesel engine in this field in America are discussed in a subsequent chapter dealing with fuels suitable for the Diesel engine.

#### METHODS OF REMOVING FUEL NEEDLE.

To remove the fuel needle from the valve it is necessary in some constructions to remove the nut holding the spring bonnet and cap in place, as well as the two nuts that fix the position of the rocker arm through which the needle valve is actuated.



A fuel-valve construction permitting easy and rapid removal of the fuel needle or the upper part of the valve without having to remove the actuating lever, is shown in figure 26. The rocking arm *a* presses upward against a small auxiliary lever *b*, pivoted on a small horizontal joint *c*, to which is secured another parallel lever *d*, which is mounted in the interior of the valve body and transmits the lifting motion to the valve needle. One end of the small journal on which the two parallel levers are fulcrumed is of reduced section and is held fixed by a nut, its function being that of a torsional spring. The opposite end is closed by a stuffing box.

A construction that facilitates the removal of the fuel needle is of decided advantage, as the valve needle has to be removed when it is ground and frequent grinding is necessary. Fuel needles have to be ground weekly or monthly, depending on the quality of the fuel oil. The use of oils with high ash content necessitates more frequent grinding of the needle. The needle is turned back and forth in its seat, a small quantity of emery dust and oil being used. A number of constructions other than that described accomplish the same objects by different means.

Connections of the copper or seamless-steel tubing used for carrying the air and fuel under high pressure are made by joining the ends of the pipes to a copper shank terminating in a cone which fits into a tapered seat machined out of the valve body. A steel gland nut slipped over the copper shank is pressed against the cone to seal the seated connection.

To seal the air chamber of the fuel valve and prevent leakage of air around the valve needle, lead or Babbitt-metal shavings mixed with flaked graphite are used as packing material, secured by appropriate glands. A series of labyrinth grooves on the valve needle constitutes an added precaution against serious leaks.

#### METHODS OF ACTUATING VALVES.

The time of opening and closing of the different valves of Diesel engines must be accurately controlled; they are therefore not self-acting but are operated mechanically.

#### SYSTEM OF ROCKING LEVERS AND CAMS.

The usual method of actuating the valves of vertical engines is through a system of rocking levers and cams; the latter are mounted on a horizontal shaft near the top of the cylinders. The cam shaft is driven through a set of helical gears, running in oil, by a vertical shaft, which in turn is driven through another set of helical gears from the engine shaft. The vertical shaft has the same speed as the main shaft; the cam shaft runs at half the speed of the main shaft in engines having a four-stroke cycle and at the same speed as the main shaft

in engines having a two-stroke cycle. The governor, usually of the through-shaft type, is mounted on the vertical shaft, as this has the higher speed. This arrangement permits the use of a smaller governor of the standard type. Governors for horizontal engines are driven from the lay shaft, which also operates the valve gear. This shaft runs at half the speed of the engine shaft in engines having a four-stroke cycle. The governor is therefore speeded up through a set of spur gears, so that a smaller size can be used.

The usual method of actuating the valve gears of four-stroke engines is shown in figure 27. In the figure,  $a_1$ ,  $b_1$ ,  $c_1$ , and  $d_1$  represent the cams through which the corresponding rocking levers  $a$ ,  $b$ ,  $c$ , and  $d$ , fulcrumed on the short shaft  $g$ , are operated. Lever  $a$  governs the air-intake valve,  $b$  the fuel valve,  $c$  the starting valve, and  $d$  the exhaust valve.

Levers  $a$  and  $d$  are operated with shaft  $g$  as a fulcrum, with an unchanging center, levers  $b$  and  $c$  are carried on an eccentric bushing movable around shaft  $g$ . Contact between cams and levers is made through rollers, which for the air, starting, and exhaust valves are on the front, and for the fuel valve on the back of the cam shaft  $e$ . The arms  $a$ ,  $c$ , and  $d$  thus move downward and force open the corresponding valves, whereas arm  $b$  has a reversed motion to lift the needle valve. The extent and the duration of the valve lifts are determined by the degree of eccentricity and the length of arc of the cam profile, allowance being made with a small clearance between

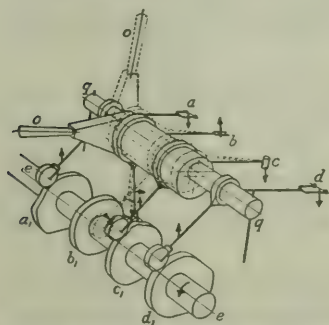


FIGURE 27.—Usual method of actuating valve gears of Diesel engines having four-stroke cycle.

the rollers and the cams during the time the valves are seated (closed). Direction of motion of the different levers is indicated in figure 27 by arrows.

By mounting the rocking arms  $b$  and  $c$  on an eccentric bushing movable around the shaft  $g$ , the following movements are made possible (fig. 28):

1. The engine is at rest, with hand lever  $o$  halfway between the vertical and the horizontal position; rocking arm  $e$ , actuating the starting valve and rocking arm  $c$ , actuating the fuel valve, are removed from contact with the cam. The starting and fuel valves are thus both disengaged. (C, fig. 28.)

2. Handle  $o$  is turned to a horizontal position (A, fig. 28), bringing roller  $e$  in contact with the cam and engaging the starting valve, roller  $c$  being still further removed and keeping the fuel valve disengaged. Opening the valve of the air receiver will admit air through the starting valve and start the engine.

3. When the engine speed required to produce the necessary compression for ignition is obtained, lever *o* is turned to the vertical position (B, fig. 28). Roller *c* is brought into contact with the cam and the fuel valve is engaged, the roller *e* (starting valve) being disengaged the same instant. This is the running position. To stop the engine, the lever *o* is moved into position shown in C, figure 28.

Before these movements are started it is of course necessary to have at least one piston past its top center. Hence the engines are provided with a barring mechanism for turning the cranks into starting position. It is also necessary to pump by hand or by other means to force a small quantity of fuel oil into the pipes leading to the fuel valves, so that the engine will fire quickly.

When the air receivers are placed on the engine-room floor separate from the engine, the operator, if only one is present, will have to mount the engine platform for making the different valve shifts, as well as step down to open or close the valve of the air receiver.

To make possible the performance of this work from the engine-room floor, including the simultaneous engaging or disengaging of all of the starting valves or fuel valves, a system of auxiliary cams is used, controlled centrally from the engine-room floor. This system also permits holding the exhaust valves open during a part of the compression stroke to save compressed air in starting. By shifting the auxiliary cams out of action the regular cams come into play when the engine is running on fuel.

In another construction the air receivers for the air used in starting and injecting the fuel are attached to the front of the engine, being mounted sufficiently high so that the air-receiver valves can be operated and all other movements directed from the engine platform. With this arrangement the accessibility of some parts of the engine suffers somewhat.

#### AIR-OPERATED PISTON VALVES.

In another construction mechanically operated starting valves are eliminated, independent air-operated piston valves being substituted. These receive the starting air through a rotary distributor operated by the cam shaft. The rotary distributor can be con-

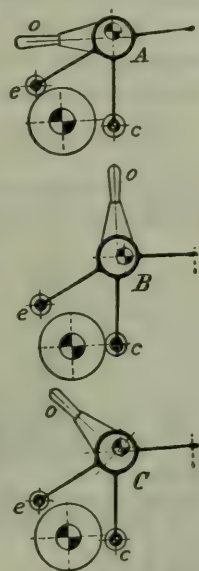


FIGURE 28.—Movements possible by special mounting of rocking arms.



nected or disconnected when the engine is running or is at rest. The fuel valves likewise are thrown in by a central control. This construction eliminates the cam and the fulcrumed lever for each starting valve used with the other types.

As a rule only one or two cylinders of a multicylinder engine are provided with starting valves, thus reducing the cost of the engine. In some engines automatic means are provided for keeping the exhaust valve open during starting, compression being avoided until the engine is well up to or past its normal speed to save compressed air in starting. Some horizontal engines have a blow-off cock in the end of the cylinder, which is kept open for a time in starting; this is also used to blow off accumulations of dirty and spent cylinder oil.

#### DETAILS OF VALVE-ACTUATING DEVICES.

The rocking-arm levers and the shaft on which they oscillate are carried by individual column stands mounted on each cylinder head. The details of such a stand are shown in figure 29. The shaft (corresponding to shaft *g*, in fig. 27), is fixed in its position with stud bolts. The rocking arms move rapidly and have to resist considerable bending moments; they should be both light and strong. They must also retain their original shape and not be subject to deformation under stress, so as to insure the accuracy of the valve motion. They are therefore made of a good quality of cast steel. The rollers are of hardened steel. The fulcrum bearing is a bronze bushing set in the rocker arm; a ball-and-socket joint insures an articulated contact between the lever and the valve-spring caps.

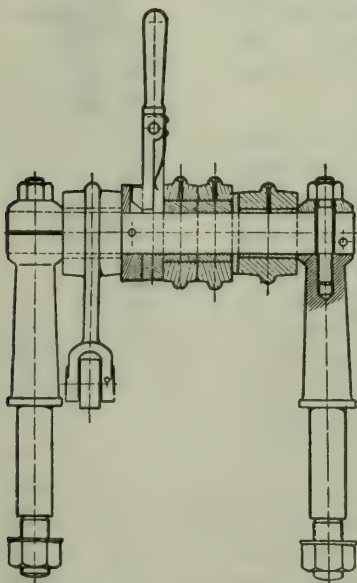


FIGURE 29.—Column stands supporting rocking-arm lever and shaft.

The cams are of cast steel. They are keyed to the cam shaft, but are usually not set definitely until the engine has been tested. The elements of a gear for operating an air or an exhaust valve are shown in figure 30.

The cam actuating the fuel valve must time the opening of the valve correctly and control the length of time the valve is open; it must therefore be carefully set to insure the best operating performance of the engine. The eccentric is a separate steel inset, se-

cured to the cam wheel with screws. Slotted holes permit the shifting of the eccentric when its correct position has been ascertained from indicator diagrams; it is then locked by small end shims (fig. 31).

As the exhaust valves soon become fouled with carbon, they have to be ground often, and it is desirable that they be readily removable without disturbing the valve-lever operating gear and the fulcrum shaft. This requirement is met by the construction shown in

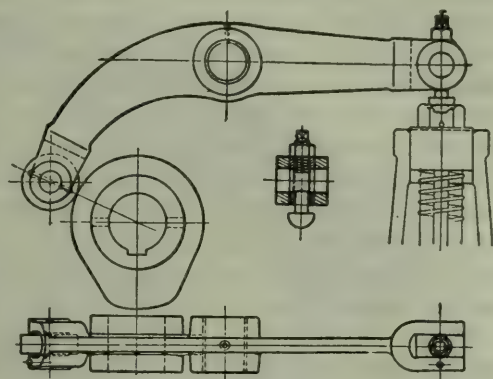


FIGURE 30.—Elements of a gear for operating an air or an exhaust valve.

Plate I, *B* (p. 20). The lever arm that engages the valve is split and the two parts are joined with two dowel pins and held together by a bolt. By removing the bolt and a part of the lever arm, the entire valve cage can be removed without disturbing the rest of the valve gear. The dowel pins insure the correct realignment of the lever arm. To remove the fuel needle, its rocking arm need not be dismantled by using a construction such as shown in figure 26 (p. 46).

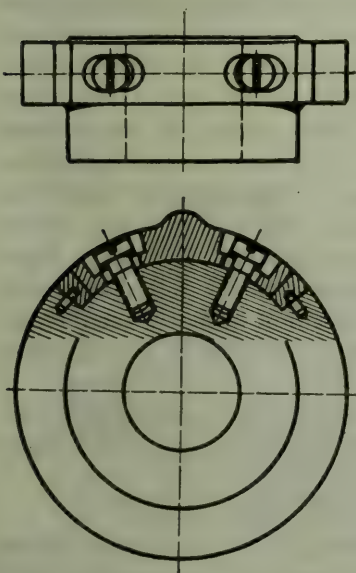


FIGURE 31.—View of cam for actuating fuel valve, showing slotted holes.

To avoid long shutdowns of the engine, a spare exhaust valve with cage is kept on hand, which is slipped into the place of the one removed. The frequency with which exhaust valves have to be removed depends greatly on the nature of the fuel burned; fuels with high ash and coke content will foul the exhaust valves more rapidly. Although engines can be operated continuously for several weeks, the usual practice is to stop an engine for about an hour weekly, and remove the exhaust valve and fuel needle of one cylinder.

The valves and needles of the other cylinders are removed in rotation, each week. The valves are removed regardless of their condition merely to keep the engine at its highest efficiency.

Another construction permitting easy removal of valve and needle is shown in Plate VI, *A*. The exhaust and air valve levers are carried by overhanging shaft ends, and can be slipped off by removing a cap bolt and washer, the washer being locked by a small pin set in a hole in the shaft end. In lieu of the two-column stand for supporting the lever shaft, a forked bracket is used, with the rocker arm for the fuel needle in the middle. The needle valve can be removed without disturbing its actuating lever, through an auxiliary system of levers, which reverse the lever-lifting motion. By this means the rocker-arm roller for the fuel needle, in common with the other rocker arms, is engaged by the cam on the front of the cam shaft. The rocker arm for the starting valve is eliminated by the use of a starting valve operated by air.

When the scavenging valves of two-cycle engines are placed in the cylinder head, the four valves are operated in pairs; and each yoke (or valve pair) is actuated by a system of double rocking arms, so fulcrumed as to form a parallelogram which serves to actuate all four valves simultaneously and to open each valve an equal amount (see fig. 17, p. 33).

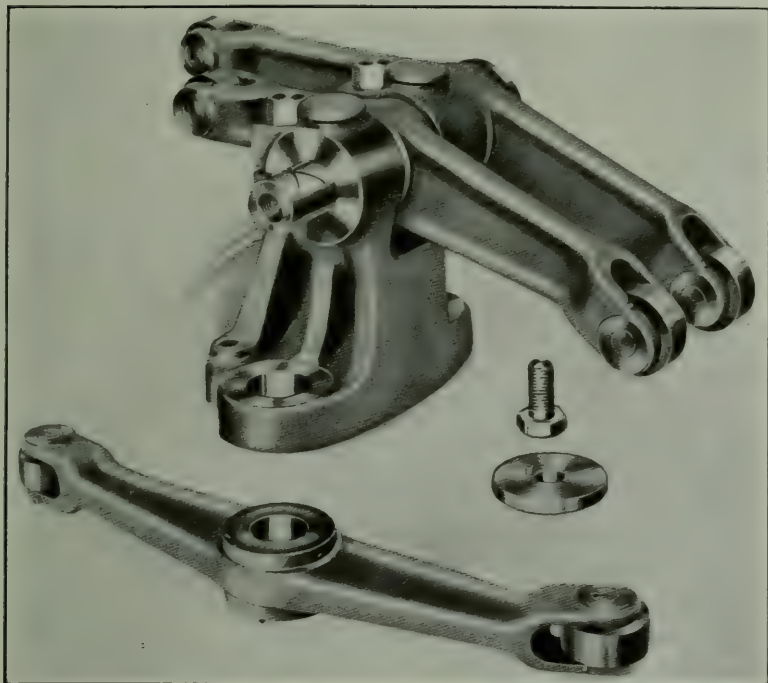
One type of cam shaft is supported by vertical bracket bearings, bolted to facings on the cylinder frames. Ring-oiling cam-shaft bearings are preferable. As the cam surfaces and rocking-arm rollers in contact with each other should be well oiled, some builders house the cam shaft for its entire length and place the cam-shaft bearings in the housing. The housing is carried by brackets cast integral with the cylinder frame and with horizontal machined surfaces. This construction insures a rigid support for the housing and correct alignment of the cam shaft. The cams run in oil, and the oil is carried to the bearings by wipers. Hinged covers allow the inspection of cams, rollers, and bearings. The covers when closed prevent spilling and flying of oil. This construction insures thorough lubrication of all rubbing surfaces, reduces to a minimum friction in the operation of the valve gear, and keeps the engine and the engine room clean. Plate VII shows a cam-shaft housing with housing covers open.

Plate II, *B* (p. 26), shows a method of driving a vertical governor shaft on a vertical engine through helical gears from the main shaft.

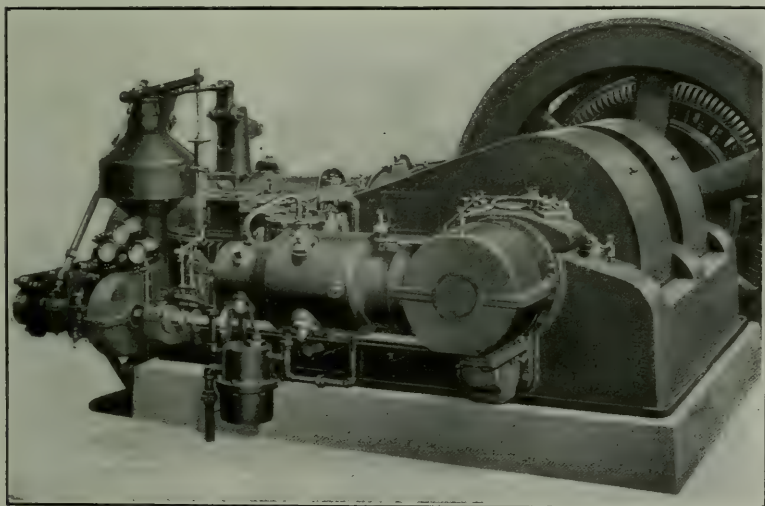
#### VALVE-ACTUATING MECHANISMS IN HORIZONTAL ENGINES.

In horizontal engines the governor shaft is driven through helical gears from the engine main shaft, and is mounted horizontally along one side of the engine frame. If the valves are placed in the end of the head, a placing corresponding to the top of the head in vertical engines, the cam shaft is mounted in brackets similar to those of vertical engines. This shaft is driven through a set of bevel gears from the governor shaft.

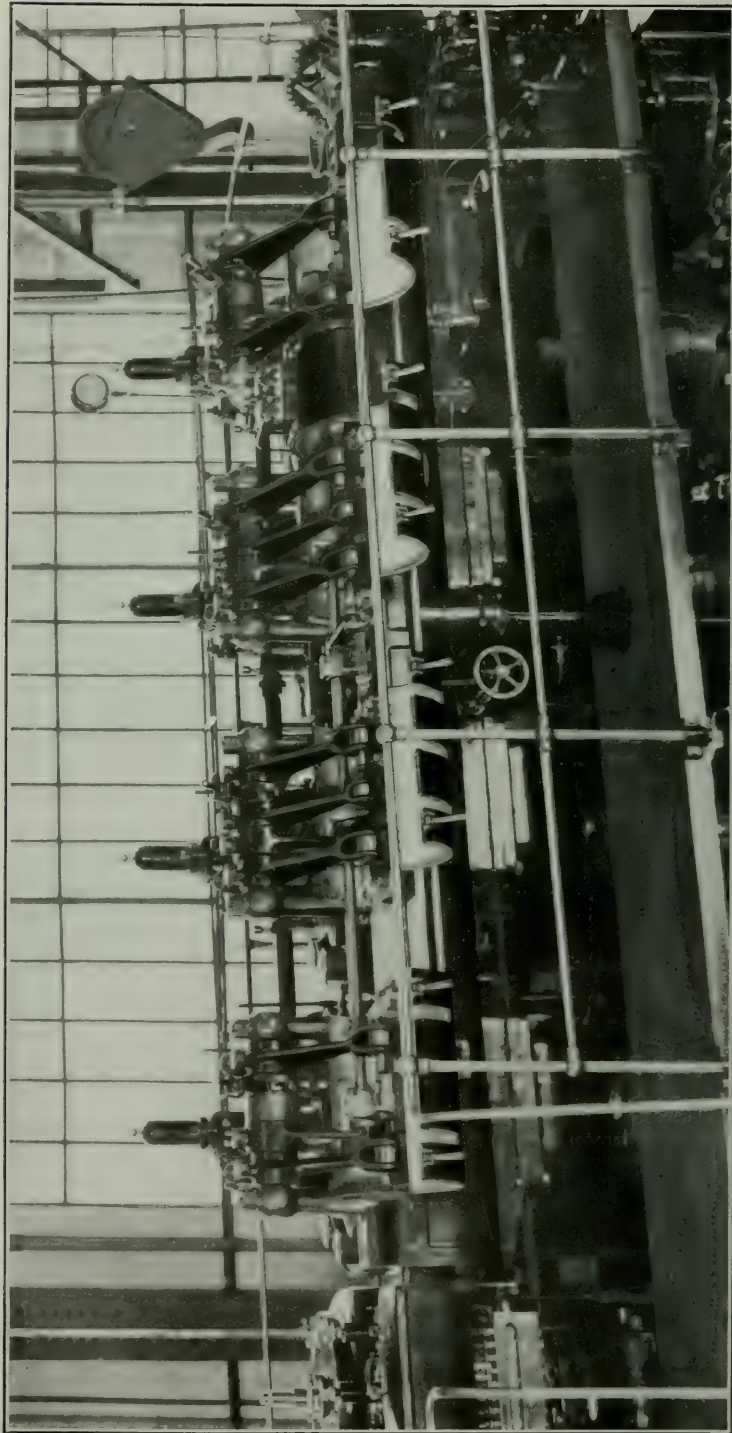




A. CONSTRUCTION OF FUEL-VALVE LEVERS PERMITTING EASY REMOVAL OF VALVE AND NEEDLE.



B. DIESEL ENGINE BUILT BY ALLIS-CHALMERS MANUFACTURING CO.



DIESEL ENGINE BUILT BY BUSCH-SULZER BROS. DIESEL ENGINE CO. SHOWS CAM-SHAFT HOUSING WITH HOUSING COVERS OPEN.

In horizontal engines having gas-engine heads with the air valves in the top and the exhaust valves in the bottom of the head, the valve levers are operated through eccentrics from the lay shaft. To accelerate the lifting of the valves and to retard their seating, cams with rolling contacts are used. The fuel valve is placed in the center of the head and is operated through a lever driven by an eccentric from the lay shaft, or a separate short shaft is used at right angles to the lay shaft and across the cylinder head, from which the fuel valve is engaged by a cam.

The starting valve or valves are operated by air and are controlled by air-distributing valves operated by a cam on the lay shaft, with which they are brought in or out of contact by a hand lever. Other engines mount the cams on an eccentric sleeve, by the shifting of which the starting valve is engaged, the exhaust valve held open during the compression stroke, and the fuel valve disengaged while the engine is started; shifting this eccentric sleeve to the operating position disengages the starting valve and engages the fuel valve and the other valves in their proper sequence. This arrangement is similar to that used in vertical engines.

The valves of horizontal two-cylinder engines are connected by a system of parallel rods, permitting all valves to be operated from one lay and governor shaft. A separate small shaft across the two cylinder heads driven through a set of helical or bevel gears operates the fuel valve by means of cams. The starting valves are operated in the manner first above described.

The time at which the different valves are opened and closed differs widely with different makes of engines.

The air valves open  $15^{\circ}$  to  $20^{\circ}$  before the piston reaches the top dead center, and close  $15^{\circ}$  to  $20^{\circ}$  past the bottom center, being open a total period of  $210^{\circ}$  to  $220^{\circ}$ .

The fuel valves open  $2^{\circ}$  to  $8^{\circ}$  before the piston reaches the top center, and close  $18^{\circ}$  to  $36^{\circ}$  after the piston has passed the top center, being open  $20^{\circ}$  to  $44^{\circ}$ .

The exhaust valves open  $25^{\circ}$  to  $45^{\circ}$  before the piston reaches the bottom center, and close  $8^{\circ}$  to  $14^{\circ}$  after the piston has passed the top center.

The angle of lead is least for slow-speed engines and greatest for high-speed marine engines of moderate power (100 to 200 horsepower; 400 to 600 revolutions per minute). This angle makes allowance for the inertia of the valves; the type of fuel valve and the properties of the liquid fuel burned also influence the timing of the fuel valve.

#### FUEL PUMPS.

The reliability of the fuel pump and its sensitiveness in responding to the slightest variations in load largely determine the satisfactory operation of the engine. The pump must respond to the governor



instantly and must furnish a supply of fuel accurately proportioned to the needs of the engine load. The permissible variations have to deal with extremely small volumes of fuel. As noted above, a fuel volume of 2 c.c. corresponds to a full-load injection for a 50-horsepower cylinder with about 80 injections per minute (160 revolutions per minute). A variation in load of 1 per cent amounts therefore to a fuel volume of only 0.02 c.c., and illustrates how seriously the regulation of an engine may be affected by minute quantities of fuel, unless the fuel pump and governor measure with great precision the fuel required.

As we have distinguished between open-nozzle and closed-nozzle fuel valves, so we may distinguish between pumps that do not have to force the fuel against the high pressure of the injection air but supply the fuel to open nozzles, and pumps that must deliver the oil against the injection air in closed-nozzle valves.

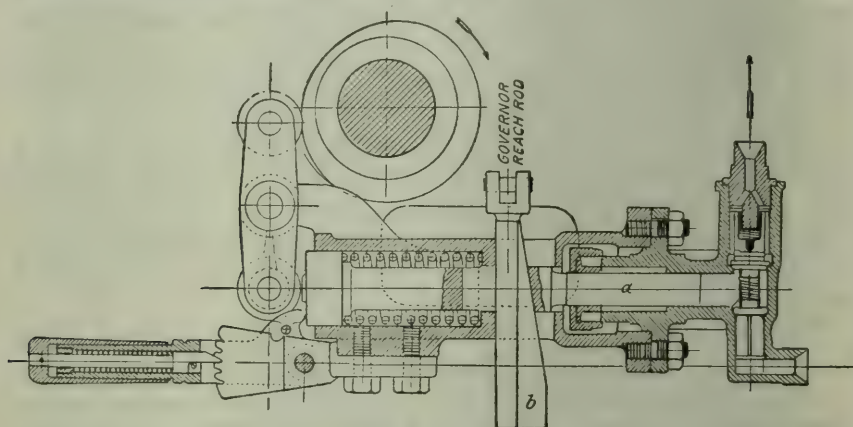


FIGURE 32.—Cross section of oil pump designed to supply oil to open nozzles without opposing air pressure.

#### PUMPS THAT SUPPLY FUEL TO OPEN NOZZLES.

Pumps of the first type have to work against little resistance—merely that due to pipe friction and the low lift from pump to fuel valve; the oil supply can therefore be regulated with close accuracy by varying the stroke of the pump plunger, which is under governor control.

Figure 32 shows a cross section of a pump of this type. The pump displacement is in proportion to the travel of the plunger *a* which is slotted; a wedge *b*, actuated by the governor moving up and down in this slot, decreasing or increasing the length of stroke and the quantity of oil delivered to the fuel valve.

The cam and the oscillating lever or rocking arm, through which the plunger is actuated, compress a spring, which on its extension reverses the direction of the plunger during the suction stroke of the pump. The plunger can also be operated by the hand lever, before the engine is started, to deliver oil to the fuel valve.

Pumps of this type are simple, work well, and are subject to little wear. When they are working under low pressure, the stuffing boxes are easily kept tight and give no trouble. They respond readily to the governor, and accurately adjust themselves to the smallest load variations.

The fuel pump shown in figure 33 is driven by an eccentric on the lay shaft. The plunger is of the differential type, the upper

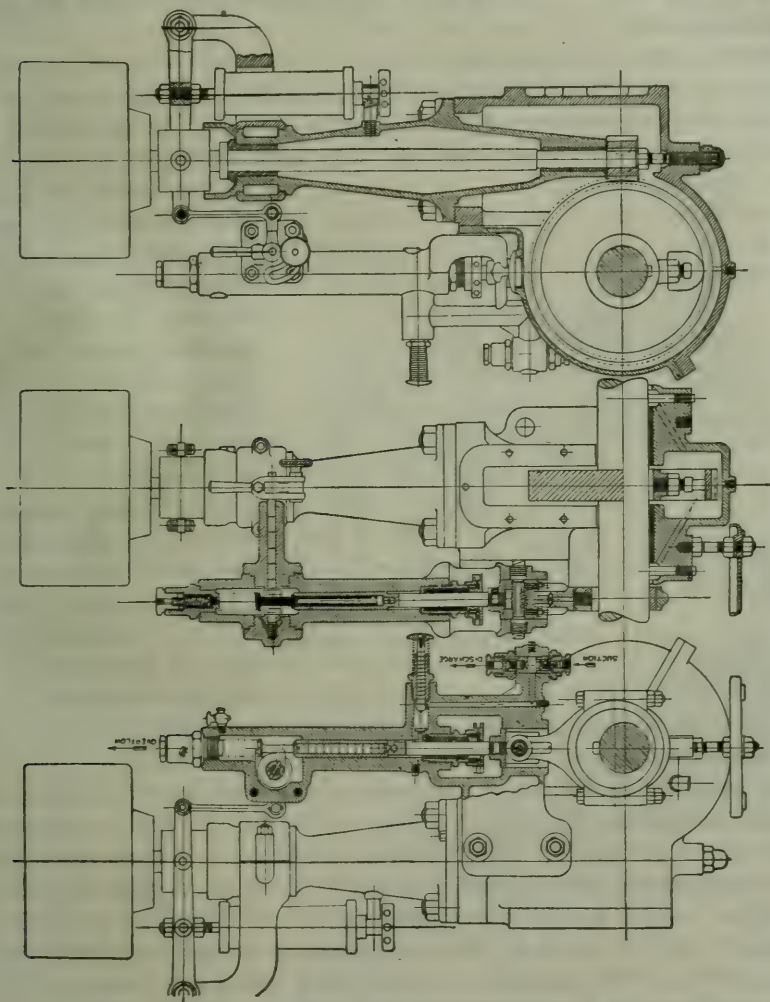


FIGURE 33.—Cross section of oil pump designed to supply oil to open nozzle.

or larger part being hollow and having seated at its upper end a cut-off valve. The plunger has a full positive stroke with each revolution of the lay shaft. The cut-off valve is under direct control of a Jahns governor. Governing is effected by permitting the cut-off valve to seat in a predetermined point in the upstroke of the plunger, thus delivering to the fuel valve a quantity of oil, correctly proportioned to the load that the engine is carrying.

## PUMPS DESIGNED TO FORCE FUEL AGAINST HIGH PRESSURE.

Pumps of the other type have to be designed to withstand safely the maximum pressure against which they have to deliver the oil, namely 1,000 pounds per square inch. They have to force the oil into the fuel-valve chamber, which is under the pressure of the injection air that varies from 600 to 1,000 pounds per square inch. Pumps of this type have to deal with extremely dense and correspondingly sluggish liquids.

For this reason no attempt is made, in pumps of this class, to place the pump plunger under governor control and to regulate the fuel-oil supply by varying the stroke of the pump plunger in proportion to the fuel required by the engine. The high pressure under which

the plunger has to work would react on the governor with each stroke, affecting its precision unfavorably and demanding a more powerful governor. There is also the possibility that air may be drawn into the plunger barrel during the suction stroke; with the minute quantities of oil to be delivered to the engine, especially at partial loads, a small air bubble may displace all the oil, and

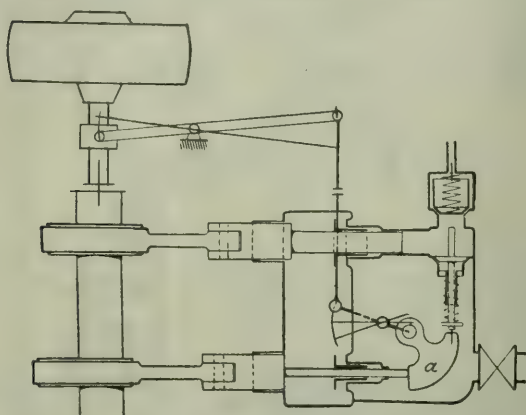


FIGURE 34.—Essential features of oil pump designed to work against pressure of injection air and having constant-stroke plunger and push rod.

as air is an elastic medium, may be compressed and expanded and seriously interfere with the functioning of the governor.

To avoid these difficulties the pump plunger works with a constant stroke, furnishing a supply of oil exceeding the maximum quantity required by the engine when working at full load. The excess fuel is by-passed, the suction valve being held open, the degree of opening being in inverse proportion to the load of the engine. The pump plunger thus delivers only the exact quantity of fuel required by the engine load. The suction valve is under governor control and is actuated through a push rod with constant or variable stroke.

The elements of a pump with a constant stroke of the plunger and the push rod are shown in figure 34. The pump is driven by eccentrics from the (vertical) governor shaft. Crank *a* is free to swing around a pin supported by one end of a fulcrumed arm, the other end of which is connected with the governor. The push rod with



constant stroke opens the suction valve by lifting crank *a* around its point of support; the magnitude of the lift is controlled by the extent that *a* is lifted under the influence of the governor. The lower the load the greater is the lift of *a* and, correspondingly, of the suction valve, with a greater quantity of the oil by-passed. Although the stroke of the push rod remains always constant, the surface of crank *a* in contact with the end of the push rod slides past it up or down when being lifted or lowered by the governor.

In other pumps the push rod has a constant stroke, but the starting point of its travel is influenced by the governor, and it subjects the lift of the suction valve to the same variation in travel.

A pump comprising all the elements of the one shown in figure 34, but with a variable-stroke push rod, has crank *a* swinging around a fixed instead of a movable center. The push rod is operated by a variable eccentric device connected with a through-shaft spring governor which shifts the eccentric device relative to the shaft center, and thus alters the stroke of the push rod. The changed lift of crank *a* affects the lift of the suction valve correspondingly.

Pumps of this type are usually provided with a hand crank attached to crank *a*, which permits the lifting of the suction valve to its extreme position and the lifting of the discharge valve sufficiently off its seat to let the oil flow by gravity into the pump chamber and the fuel-valve oil-delivery pipe from the oil-storage tank (which is higher than the engine) by opening a cock connecting the pump chamber with the atmosphere. When the suction valve is held wide open, the engine is stopped. These pumps have been developed to a high degree of perfection, and notwithstanding the severe conditions under which they have to operate they are highly reliable.

Pumps may be vertical or horizontal; the former are used if operated from the cam shaft, although pumps with a variable stroke of the plunger are usually of the horizontal type (fig. 32). Pumps driven from the vertical governor shaft of vertical engines are usually horizontal (fig. 34).

#### NUMBER OF PUMPS REQUIRED.

Practice regarding the number of fuel pumps supplied with multi-cylinder engines differs, being often decided by a desire to reduce manufacturing costs.

When only one pump is supplied, the fuel has to be divided and proportioned equally to the different cylinders. This is done with small steel diaphragms, the size of the holes being determined experimentally, allowance having to be made for the resistance in the different pipe branches leading to each fuel valve. This method is not entirely satisfactory, especially where a high degree of regulation is desired, as in synchronizing. With a decrease in load, some of the

cylinders will receive an excess of fuel, the supply being proportioned to the load before the governor acted. If one or more of the branches is obstructed, the active cylinder will be overloaded, and if any of the fuel valves or the fuel passages to them leak the active cylinders will be underloaded.

In pumps that are driven by the vertical governor shaft, which, in an engine having a four-stroke cycle, makes double the number of revolutions of the cam shaft, the plunger delivers the fuel in two parts for any one fuel charge, corresponding to one revolution of the cam shaft. The governor, by acting between the two plunger strokes, can therefore adjust the second part of the fuel charge to the new engine load, counteracting somewhat the defect mentioned. As a result of the desire to improve the regulation further, individual pumps are often built as multiplunger pumps, one plunger being provided for each fuel valve (each cylinder). As the plungers are driven simultaneously by one eccentric device, they divide the fuel effectively, but the governor continues to influence only the combined quantity of fuel delivered by all the plungers and not of that delivered by each plunger successively.

To avoid these defects many builders provide an independent pump for each cylinder, each pump being separately under the influence of the governor. The cams or eccentrics actuating these pumps are mounted on the cam shaft (or the governor shaft) with different leading angles, corresponding to the sequence at which the different fuel valves act, and each pump delivers the oil just before the fuel valve it serves opens. Engines so equipped have the most effective regulation.

### AIR COMPRESSORS AND AIR RECEIVERS.

#### GENERAL DETAILS OF AIR COMPRESSORS.

The air required for starting the engine and for atomizing and injecting the fuel is furnished by an air compressor. On account of the high pressure necessary, the air is compressed in two or three stages. Ample cooling of the compressor cylinders and of the air in intercoolers between stages is not merely desirable to approach isothermal compression and reduce the power needed for compression, but is necessary to avoid accidents that may wreck the compressor, with possible injury to the operators. With insufficient cooling of air and excessive use of cylinder-lubricating oil or of an unstable oil, the oil vapor may form an explosive mixture with the heated air and be ignited upon compression. Three-stage compression is therefore much to be preferred, as it permits compressing the air more gradually, and the lower compression ratios avoid excessive terminal temperatures; it also permits cooling the air more thoroughly in intercoolers before it is passed from one stage to the succeeding stage,

The air leaving the last stage of the compressor should be passed through an aftercooler and an oil separator before it is stored in the air receivers. This step is taken not so much to prevent a drop in pressure in the injection air bottle with the contraction of the heated air on cooling as to avoid explosions of the heated air charged with oil vapor. Regrettable accidents have happened from failure to provide intercoolers and aftercoolers. Not only pipes but air receivers as well, have burst. Flame passing through the air pipe and into the air receiver charged with oil vapor has ignited this vapor and blown up the receiver. The aftercooler cools the air, and the oil-and-water vapors are condensed in the separator, and from time to time are drained.

The air compressor should be lubricated thoroughly but without the use of an excessive quantity of oil. The oil used should have a paraffin base, should be of great stability at high temperature, and should have high flash and burning points.

The air compressor is usually driven direct from the main shaft of the engine. On multicylinder engines with closed crank case the compressor usually has the appearance of an additional cylinder, the compressor cylinders and pipe-coil air coolers being surrounded with a mantle forming a water space.

On some engines, especially large units, more than one compressor is used to avoid the large size necessary with only a single compressor. Moreover greater accessibility and reliability are obtained, and the work of compression is divided into a larger number of smaller absolute impulses.

Compressors may be vertical, horizontal, or inclined for both vertical and horizontal engines. They are made of cast iron, with water-jacketed cylinders, or are surrounded by a mantle which forms a large water space and in which are housed the air-cooling coils. More systematic cooling can be done by using water-jacketed cylinders and independent intercoolers. Also, these then become more accessible. The cylinders are stepped down to correspond to the number of stages; likewise the pistons, which are of cast iron. The high-pressure end is usually too small in diameter to permit extending the snap rings sufficiently to slip them over the piston. They are then held between spacing rings and locked with a nut in the end of the piston.

The valves are metal poppet or disk valves. The discharge and intake valves between stages are preferably housed together in one cage, of a construction that facilitates their removal and replacement. The high pressure demands small clearance spaces; the interior of the cylinder-head end should therefore be free of pockets or dead spaces, and the valve seats should conform closely to the inner surface of the cylinder; the piston likewise should conform to the shape of the cylinder head to permit its close approach to the head. Each



intercooler should be fitted with an oil separator, so that excess lubricating oil and condensed water may be separated and drained and not allowed to pass on to the next stage.

A blow-off (safety) valve should be placed in the high-pressure air pipe, leading to the air receiver, directly back of the high-pressure end of the compressor (past its discharge valve), to prevent the bursting of the pipe as the result of an obstruction. The necessity of an aftercooler and an oil separator has already been mentioned.

Compressor troubles are probably responsible for a large proportion of shutdowns in the operation of the Diesel engine. Particular care should be used in compressor construction to make all the parts as simple as possible and still retain the maximum efficiency and reliability. Valves should be readily interchangeable, and the material and workmanship of the highest order.

#### AIR RECEIVERS FOR AIR COMPRESSOR.

The air for the compressor is usually stored in three air receivers, two large ones for the air used in starting the engine and a smaller one for the injection of air. These receivers are made of seamless drawn steel, and to avoid heavy walls and excessive weight are relatively long and of small diameter ( $L/D=4$  to 8). They are interconnected with a system of pipes controlled by valves, so that either of the large bottles may be replenished with air from the injection-air bottle, which is supplied from the air compressor. The valve bodies are machined out of solid forged-steel blocks. The valve bodies are connected to the air pipes by copper cones and steel gland nuts. The valves are controlled by large (8-inch) handwheels.

One of the large air bottles is used for storing a reserve supply of air under a pressure of 1,000 pounds per square inch. Air is drawn from the other when the engine is started. The latter bottle is refilled as soon as the engine is in operation. The air compressor is designed amply large to replenish, in 15 to 20 minutes running, the air so used, without drawing on that used for injecting the fuel with the engine at full load. Manometers are provided to indicate the pressure in the large bottles and the injection-air bottle and the intermediate pressure of the compressor. At least one of the large air receivers is filled with air at the manufacturer's works and shipped with the engine to be used in starting it for the first time. Before the engine is started at any time the operator should convince himself that every part of it is in operating condition; should he fail to start it by turning on the starting air, he should close the air valve at once and locate the cause of failure rather than make a number of attempts and waste the compressed air.

If all the air has been used before the operator has succeeded in starting the engine, which operates the compressor for producing a fresh supply, compressed carbon dioxide (carbolic-acid gas) can be

used in lieu of compressed air. To prevent the freezing of the gas when it is drawn, the top part of the container may be warmed by wrapping cloths soaked in hot water around it. The container should not be heated by such means as torches, as there is great danger of bursting it from overheating the gas.

The size and the number of air receivers vary with the size of the installation. In power plants using a number of engines it is well to interconnect the air receivers of the different engines, so that any one can be drawn upon in emergencies for any of the engines. Small engines need a relatively greater air-storage capacity per horsepower than do large engines. The capacity required varies from 15 to 20 gallons per 100 horsepower for the injection-air bottle and from 60 to 200 gallons per 100 horsepower for the starting-air receivers.

#### REGULATION OF AIR SUPPLY.

The air supply to the fuel valves is regulated by throttling the discharge valve on the injection-air bottle, and the desired pressure, indicated by a manometer on the injection-air bottle, is maintained by correspondingly throttling the air going to the air compressor at the air intake.

Each bottle is provided with a cock and a drain pipe reaching to the bottom of the bottle to drain it of accumulated condensed water and oil.

For starting some engines low-pressure air is used (100 to 250 pounds per square inch), which is stored in the standard type of sheet-steel, riveted air receivers. The receivers are filled with air drawn from the injection-air bottle and supplied by the air compressor. In other engines the air receivers are eliminated altogether and a seamless steel pipe is used between the compressor and the fuel valves, the pipe taking the place of the injection-air bottle. A separate small compressor driven by a gasoline engine supplies the starting air. This practice lowers the first cost of the engine, but the absence of any air reserve makes it necessary that the compressor furnishing the injection air be of unfailing reliability. With the opening and closing of the fuel valves and the lack of sufficient receiver capacity the pressure fluctuates, and with an obstruction in the pipe the pressure may rise abruptly and burst the pipe, as even the precautionary blow-off valves can not be depended upon to act unfaithfully. The presence of a gasoline engine with its highly inflammable fuel introduces a fire risk that is absent in installations that derive the starting air from the engine-driven compressor and have in the engine room only a small supply of fuel oil with a high flash point. This advantage of the Diesel engine should not be appraised too lightly in many industrial plants, such as textile and flour mills.

## MATERIAL USED IN AIR PIPES.

The injection-air and fuel pipes subjected to the fuel-pump pressure are made either of seamless-drawn copper or steel. Copper pipes are preferable on account of their greater ductility and uniform strength, even though they have a lower tensile strength than steel pipes. Seamless drawn-steel pipes have failed, owing to local imperfections, where copper pipes have satisfactorily stood the test; but no receiver system or pipes are entirely safe against rupture if the air discharged from the air compressor is not first cooled to remove any entrained oil. For conducting the air used in starting the engine a seamless drawn-steel pipe is used; this pipe is of larger diameter and, as the pressure of air used for starting should not exceed 500 pounds per square inch when admitted to the engine cylinder, is safe. This air is usually at atmospheric temperature and the danger from ignition of explosive mixtures of air and oil vapors is absent.

For circulating cooling water and supplying fuel oil to the fuel-storage tank in the engine room standard black or galvanized-iron pipes are used.

## SOURCE AND VOLUME OF AIR USED.

The air for the engine is taken directly from the engine room or from the outside. If the air is taken from the engine room, the customary method is to bolt to the facing of the cylinder head a cast-iron elbow, connecting this with the intake to the air valves; the elbow points downward and is connected with a short length of steel pipe (one elbow and one pipe for each cylinder). Its end is closed with a cap. The air enters through a series of long slits cut into the walls of the pipe. This arrangement lessens the noise and prevents foreign bodies from being drawn into the engine.

As the volume of air drawn into the engine is considerable—about 635 cubic feet per minute for a 100-horsepower engine having a four-stroke cycle and about 850 cubic feet per minute for an engine having a two-stroke cycle—especially in large installations, means of admitting an excess of air to the engine room must be provided. However, the noise will probably be considerable and the windows will vibrate and shake under the continuous waves produced by the periodic air displacements. It is therefore a better plan in large installations to draw the air from the outside through large conduits built in the foundation and connecting with the different air intakes to the valves.

Two-stroke engines are provided with air through the air (scavenging) pump, so it is necessary merely to have the pump intake connected with the air canal.



Wherever the air is likely to be contaminated with dust, it should be filtered to prevent excessive cylinder wear and the shortening of the life of the engine.

#### EXHAUST PIPES.

The exhaust pipes should be of cast iron as steel pipes corrode rapidly, especially when the fuel oil contains an appreciable quantity of sulphur. The sulphur burns to sulphur dioxide, which may be oxidized to the trioxide in the engine cylinder, and combine with the water vapor of combustion to form sulphurous or sulphuric acid. The exhaust gases should never be cooled to the condensing point of water, which would cause corrosion. As the exhaust gases issuing from the engine are hot ( $600^{\circ}$  to  $1,000^{\circ}$  F.), the exhaust pipes are jacketed and water cooled in the proximity of the engine to make work around the engine bearable to the attendants and to prevent burns.

It is customary to provide a test cock in the exhaust pipe near the cylinder head. By holding a piece of white paper over this cock determination can easily be made as to whether combustion is perfect or incomplete. The exhaust should be colorless.

#### STORAGE OF FUEL OIL.

The storing of the fuel oil does not call for any departure from the customary practice in steam-power plants. Closed steel drums set in individual closed arched concrete cellars are somewhat high in first cost, but combine the advantages of low fire risk, easy detection of leaks, and prevention of evaporation loss. They can be filled direct from tank cars. As Diesel plants use about one-third the quantity of fuel oil consumed by efficient steam plants, the storage capacity can be proportionately reduced. The rooms in which the fuel tanks are housed should be properly ventilated to prevent the accumulation of any oil vapor that may escape. Care should be taken not to draw the oil from too near the bottom of the oil tank, to prevent sediment entering the oil-supply pipes.

In the engine room are one or more small fuel-supply tanks, usually of a capacity to run the engine for half a day. In small installations, a hand-operated fuel pump is used to refill the small fuel tank with fuel oil from the main fuel-oil-storage tanks. In larger installations a motor-driven pump is used. The small fuel tanks are supplied with a glass gage to indicate the fuel level. Whenever the viscosity of the oil is such that it will not flow readily through the pipes at ordinary temperatures, means to heat the oil to increase its fluidity sufficiently have to be provided. The usual method is to pass the hot jacket water through pipe coils in the small fuel tanks, or to have these tanks provided with water jackets, through which the hot water from the

engine jacket is passed. Provision then has to be made for another smaller fuel tank, in which gas oil (ignition oil) for starting the engine is stored: this is switched on when the engine is to be stopped for any length of time. The fuel supplied to the pipe leading to the engine fuel pump or pumps is controlled from either supply tank by a three-way cock.

When the fuel oil contains sand or other foreign matter, small filter tanks are provided; the fuel oil from the small fuel-supply tank flows through these filter tanks into the main fuel-supply pipe to the engine fuel pump through two branches (one from each filter tank) controlled by a three-way cock. This arrangement permits switching one tank off the circuit for cleaning while the other continues to supply the engine with fuel. As an additional safeguard, filter plugs are placed ahead of the intake valves of the fuel pumps.

The fuel tanks are supported on brackets or a platform on the wall opposite the engine and high enough above it for the fuel to flow by gravity to the engine fuel pumps. Usually each engine is supplied from its own small fuel-supply tank.

#### COOLING WATER.

For cooling the compressor and the engine a constant flow of water, adjusted to the engine requirements, is desirable. This is best supplied from a constant-head tank, which should be at least 20 feet above the water inlet to the engine, so that the head of water will overcome any vapor tension of the steam formed within the engine jackets.

In some engines the water is first passed through the aftercooler and the intercooler and the water jackets of the compressor, then through the engine-cylinder jackets, and last through the head; in others independent water connections are used, with branches from the main supply pipe to the compressor and the different engine cylinders. Although the quantity of water required for cooling the engine is so much larger than that needed by the compressor that it is not warmed appreciably by passing through the latter first, an independent water supply to the compressor has much to recommend it, affording easier adjustment of the supply of cooling water to the different cylinders and heads. Separate discharges for the water from the cylinders and from the cylinder head should be provided, each with its own thermometer, so that the temperature in a cylinder or in the head can be controlled. The discharge pipes should be provided with valves, to regulate the discharge, as the water is under pressure.

In large installations, all the water-discharge pipes are brought to a central discharge point and mounted on a switchboard; a cock regulates the overflow, and a thermometer, mounted on the board above each respective discharge, indicates the temperature. A master valve controls the main supply to the engine. When the

engine is to be shut down, only the master valve is closed, the other valves being left in adjustment.

Some care must be used in adjusting the supply of water to the engine needs, and an excess should be avoided, as it will cool the cylinders too much; they will contract, whereas the piston, being hot, will expand. Excessive cooling may thus cause piston seizures.

When the engine is stopped, the water should be continued in circulation for some time, as the heat stored in the piston, the cylinders, and the cylinder head is considerable, being sufficient to bring the water to the boiling point, so that sudden stopping of the circulation of the water might cause the cracking of a cylinder or of the cylinder head. If the water is hard, lime or magnesia salts may be deposited and interfere with the efficient cooling of the cylinder head and the cylinders, set up internal stresses, and lead to cracked heads and liners. When the water is hard, the best policy is to use distilled water, which should be cooled before use. The loss with an efficient cooling system need not be more than 10 per cent of the volume of water needed for cooling—amounting to one-fourth to one-half gallon of water per horsepower-hour.

The volume of cooling water required varies with the size and type of the engine. Engines having a four-stroke cycle use from 2.7 to 4 gallons of cooling water a horsepower-hour, the larger volume corresponding to smaller engines. Two-stroke engines use from 5 to 6 gallons a horsepower-hour. The volume used depends on the initial and the terminal temperature of the water. The figures given are based on an initial temperature of 50° F. and a discharge temperature of 160° F. It is well to plan on not less than 5 gallons of water per horsepower-hour for an engine having a four-stroke cycle, and not less than 8 gallons for an engine having a two-stroke cycle. The capacity of the circulating pump should be 50 to 100 per cent larger, depending on the type of pump used.

#### MECHANICAL EFFICIENCY OF DIFFERENT TYPES OF DIESEL ENGINES.

The mechanical efficiency of the Diesel engine,  $E_m$ , is the net effective power developed in the engine cylinder remaining after the power absorbed by the moving parts of the engine and by frictional resistance has been deducted. The air compressor, furnishing the injection air and usually driven direct from the engine, consumes considerable of the total power of the engine (7 to 15 per cent) and this loss must also be deducted to ascertain the net effective output of the engine. The mechanical efficiency is expressed as the ratio between the effective power of the engine as measured by a brake on the engine shaft (brake horsepower) and the indicated power as determined from indicator diagrams:  $E_m = \frac{\text{BHP}}{\text{IHP}}$ .



The mechanical efficiency of a Diesel engine is influenced by numerous factors, such as the type and the size of engine, the quality of the material and workmanship, the care given to details in erecting, the lubricating system, including the quantity and the quality of lubricating oil used, the engine speed, and the volume of cooling

water used. If too much cooling water is used, frictional resistance due to cylinder contraction may be greatly increased. It is always higher in new engines, until the different moving parts have worn the rubbing surfaces smooth.

As the internal power and friction of an engine are nearly constant regardless of load, the mechanical efficiency decreases with a decrease in the engine load.

The mechanical efficiency of engines having a two-stroke cycle is lower than that of engines having a four-stroke cycle as, in addition to the power required by the injection air compressor, there is that required by the scavenging pump.

The mechanical efficiencies of four-stroke engines at full load vary from 75 to 82 per cent, 80 per cent being usual for high-grade, low-speed engines of medium and large powers. The engine efficiency, exclusive of the air compressor, is 85 to 90 per cent.

The mechanical efficiency of engines having a two-stroke cycle seldom exceeds 70 per cent and may be as low as 65 per cent in high-speed engines. The

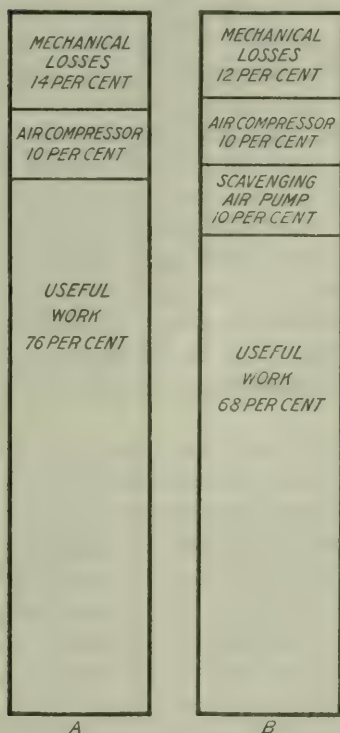


FIGURE 35.—Distribution of power losses in two types of Diesel oil engines: *A*, engine having four-stroke cycle; *B*, engine having two-stroke cycle.

distribution of power losses in two-stroke and four-stroke engines is shown in figure 35.

### THERMAL EFFICIENCIES.

The thermal efficiency of a Diesel engine is the ratio between the equivalent in heat units of 1 horsepower and the number of heat units actually consumed by the engine in developing 1 horsepower. If based on the indicated horsepower, it is the indicated thermal efficiency,  $E_{ti}$ ; if on the brake horsepower, it is effective thermal efficiency,  $E_{te}$ .

$$E_{ti} = \frac{550 \times 3600}{778} \times \frac{\text{IHP}}{\text{WH}} = 2545 \frac{\text{IHP}}{\text{WH}}$$

$$E_{te} = \frac{550 \times 3600}{778} \times \frac{\text{BHP}}{\text{WH}} = 2545 \frac{\text{BHP}}{\text{WH}}$$

In the foregoing expressions, 550 foot-pounds per second = 1 horsepower; 778 foot-pounds is the mechanical equivalent of 1 British thermal unit;  $W$  is the weight in pounds of the fuel consumed during 1 hour;  $H$  is the heating value of the fuel in British thermal units per pound.

If  $WH$  is the fuel consumption per horsepower hour, expressed in British thermal units, then  $E_t = \frac{2545}{WH}$ .

The thermal efficiency depends chiefly on the thermodynamic cycle of the Diesel engine, and is affected by the compression ratio (ratio between total cylinder volume and clearance volume at the end of compression) as well as the cut-off ratio (ratio between cylinder volume at time fuel valve closes and volume at inner dead center of piston or clearance volume).

The indicated thermal efficiency increases with a decrease in the cut-off ratio which contributes to the phenomenal economy of the Diesel engine at fractional loads, the fuel consumption per horsepower-hour remaining nearly constant between full and three-fourths load, and increasing only slightly at one-half load. The ignition of the fuel could be effected at lower pressures, but high compression is essential to high engine economy in Diesel engines.

The mechanical efficiency of the engine naturally influences its fuel economy (thermal efficiency) also, but to a minor degree.

The indicated thermal efficiency of the Diesel engine having a four-stroke cycle varies from 45 per cent at full load to 47 per cent at half load, and the effective thermal efficiency from 37 per cent at full load to 30 per cent at half load, which represents the best practice. As regards engines having a two-stroke cycle, the figures are 10 to 15 per cent lower.

#### VOLUMETRIC EFFICIENCIES.

The volumetric efficiency ( $E_v$ ) is the ratio between the weight of a cylinder full of air at the completion of the suction stroke and the weight of a similar volume at standard temperature and pressure. It can be determined by measuring the partial pressure of the air during the suction stroke and dividing it by the atmospheric pressure. The construction of the engine, its piston speed, its valve gear, and temperature are factors that influence the volumetric efficiency. The volumetric efficiency of an engine having a four-stroke cycle differs from that of an engine having a two-stroke cycle. The volumetric efficiency influences the specific duty of the Diesel engine. During the suction stroke of the four-cycle engine, the air becomes somewhat rarified, so that the lower the volumetric efficiency, the lower is the weight of oxygen in a cylinder full of air. The maximum quantity of fuel that can be burned by the air (oxygen) charges is also proportionately lower with lower volumetric efficiency.

The volumetric efficiency of engines having a two-stroke cycle is generally below unity, notwithstanding the fact that the cylinders are filled with slightly compressed air, as it is not possible to scavenge or remove all the gases of combustion, which vitiate the burning power of the air charge. To determine the volumetric efficiency of engines having a two-stroke cycle, it is not sufficient to know the pressure of the air that fills the cylinder (before compression begins); this value must be multiplied by the percentage of pure air present in the total weight of gas filling the cylinder.

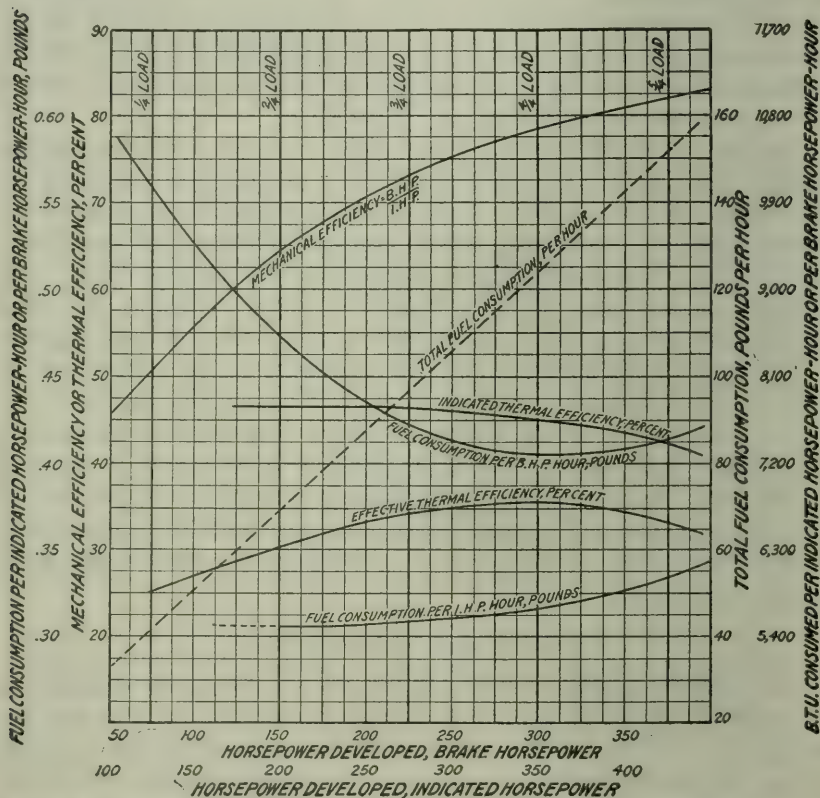


FIGURE 36.—Typical performance curves representing engine efficiencies of 300-horsepower Diesel engines.

For slow-speed four-stroke engines a volumetric efficiency of 90 per cent can be reached, which decreases to 85 per cent for high-speed engines, and for extreme speeds may be even lower. These values presuppose high-grade engines with mechanically operated valves.

Typical performance curves representing engine efficiencies of a 300-horsepower Diesel engine are shown in figure 36.

#### EFFECT OF HIGH ALTITUDES ON EFFICIENCY.

At higher altitudes the specific duty of Diesel engines decreases appreciably, owing to the lessened density of the air, which affects



the engine just as a low volumetric efficiency would. The horsepower rating of the engine decreases 3 per cent for every 1,000 feet of added altitude. Nearly 40 per cent of the power loss could be recovered by precompressing the rarified air to atmospheric or slightly higher pressure in positive-pressure blowers and filling the engine cylinders with this air. Blower equipment is considerably cheaper per horsepower of capacity than Diesel engine equipment. At high altitudes it may pay, under certain conditions, to install blower equipment for precompressing the air for engines having a four-stroke cycle. An engine having a two-stroke cycle can compress the air readily by using a larger scavenging pump.

#### CHARACTERISTICS AND USES OF HIGH-SPEED ENGINES.

Diesel engines may be classed as low-speed and high-speed engines. The former are preferred for hard, continuous duty. They have relatively low piston speed, varying from 600 to 800 feet per minute, the piston speed increasing with the power. The number of revolutions per minute varies from 250 to 150, decreasing with the size of the engine. The stroke-bore ratio varies from 1.3 to 1.9, the higher ratio being preferred for low-speed engines for hard service, although with an increase in piston speed the stroke-bore ratio decreases, likewise the number of revolutions.

High-speed engines have a piston speed of 700 to 1,000 feet per minute, a stroke-bore ratio of 1.0 to 1.3, and an engine speed of 250 to 350 revolutions per minute. The speed of engines for special purposes, as for submarines, is often increased to 500 and 600 revolutions per minute with low stroke-bore ratio to obtain a light engine of low height.

The high-speed engines are useful for central-station duty, or for reserve and stand-by units, where the load is relatively light and periodical. The high rotative speed reduces the cost of the generator, makes parallel operation easier, and produces a compact, relatively low engine. The workmanship and materials used in engines of this type have to be first-class, and the materials must be of special quality to be able to withstand the greatly increased specific duty of all the moving parts. Hence engines of this type are by no means reduced in price in proportion to their increased speed, their cost being relatively higher than that of slow-speed engines.

Some manufacturers, to reduce stock sizes, build medium-speed engines only.

The mean effective pressure should not exceed 100 pounds per square inch at full load and is preferably kept around 90 pounds for hard, continuous service. The engines are designed to carry safely momentary overloads of 20 to 25 per cent, when the pressure will go as high as 120 and 125 pounds per square inch. Engines having a two-stroke cycle, in which the number of fuel combustions is generally

double that in engines having a four-stroke cycle, are designed to operate with a lower mean effective pressure—between 65 and 70 pounds per square inch—with a margin for increasing the specific duty of the engine for short periods.

### DIESEL ENGINES MADE IN THE UNITED STATES.

The important mechanical features of the Diesel engines manufactured in the United States are briefly summarized below.

#### ALLIS-CHALMERS MANUFACTURING CO., MILWAUKEE, WIS.

This company builds a horizontal engine with gas-engine head and eccentric-driven admission and exhaust valves. The horizontal com-

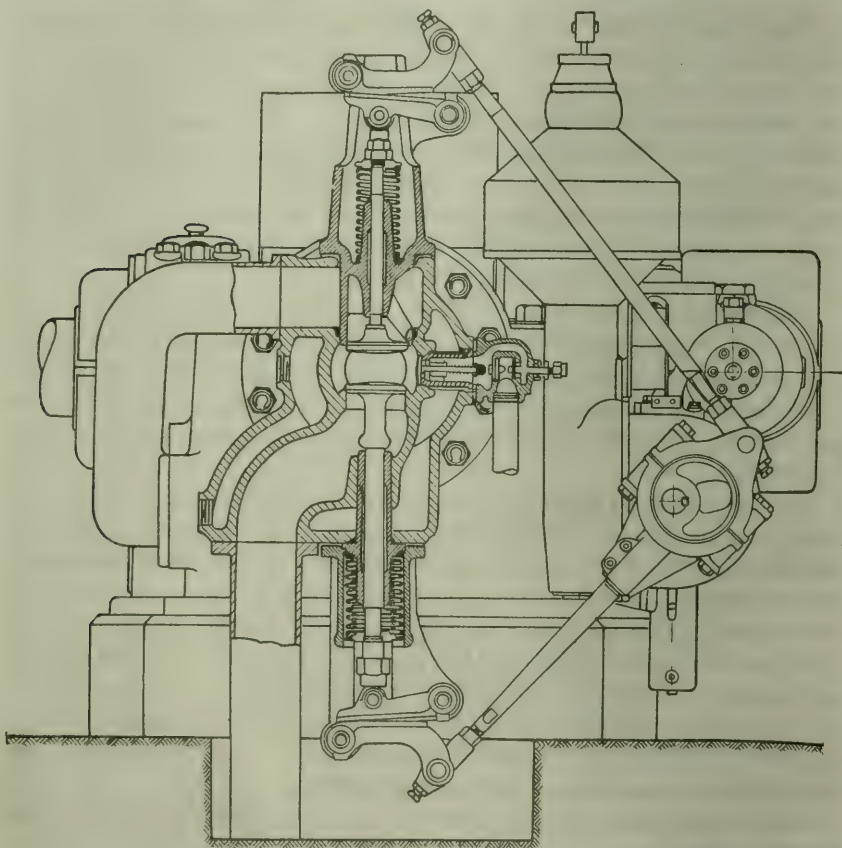
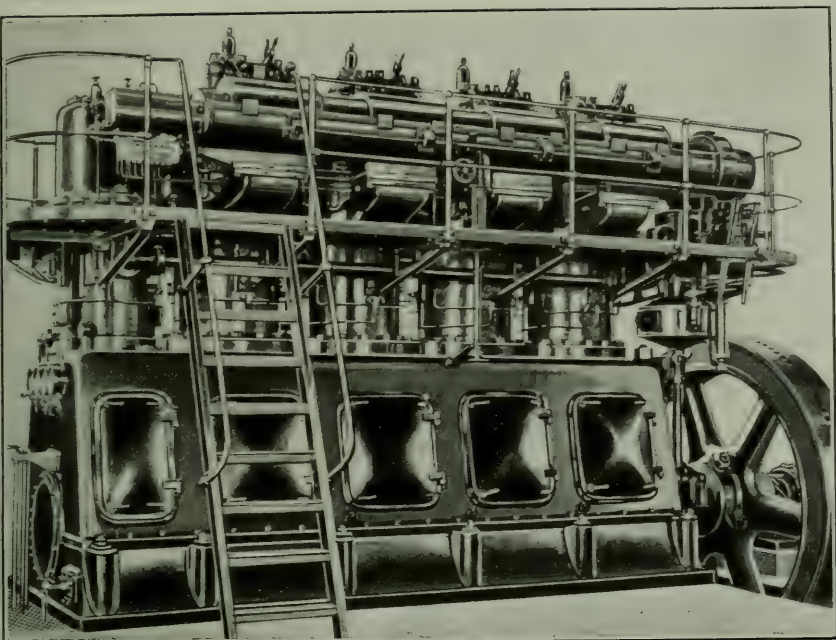
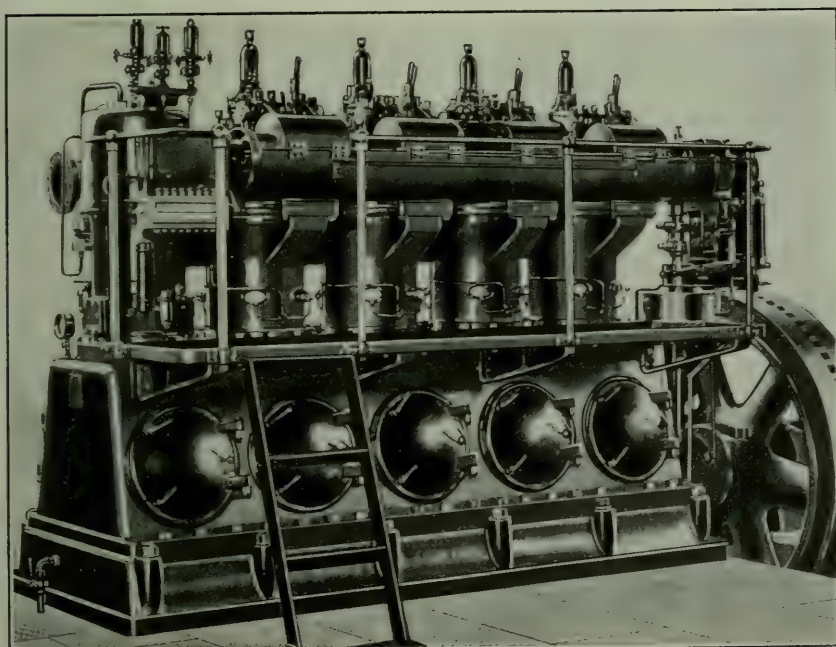


FIGURE 37. — End section of Diesel engine built by Allis-Chalmers Manufacturing Co.

pressor is mounted on the side of the engine frame. It uses the Lietzenmayer system of open fuel nozzle, and variable-stroke plunger fuel pump, which delivers the fuel in the fuel valve during the suction stroke. Figures 37 and 38 and Plate VI, B) p. 52), show views of this engine.

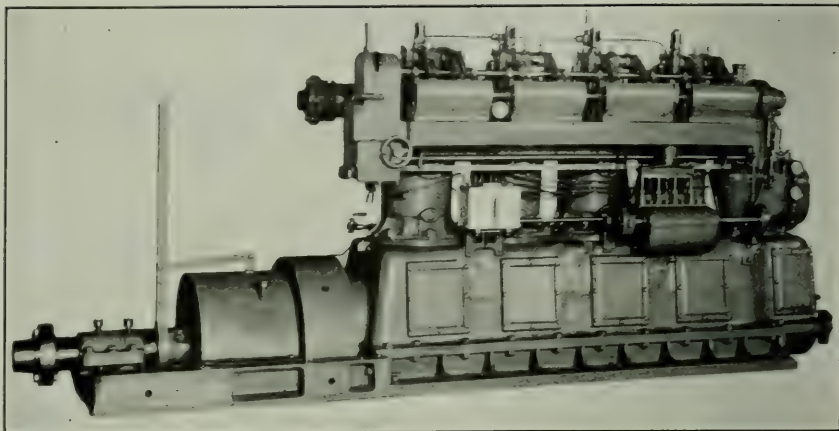


A. BUSCH-SULZER 520-BRAKE-HORSEPOWER DIESEL ENGINE.

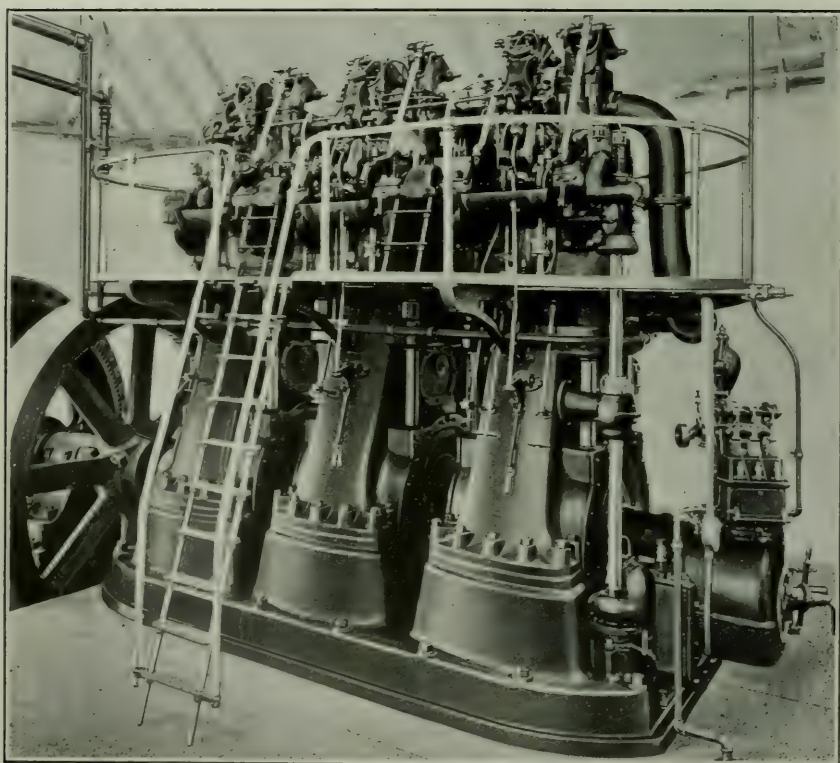


B. BUSCH-SULZER 120-BRAKE-HORSEPOWER DIESEL ENGINE.





A. DIESEL ENGINE BUILT BY FULTON MANUFACTURING CO.



B. DIESEL ENGINE BUILT BY LYONS ATLAS CO.

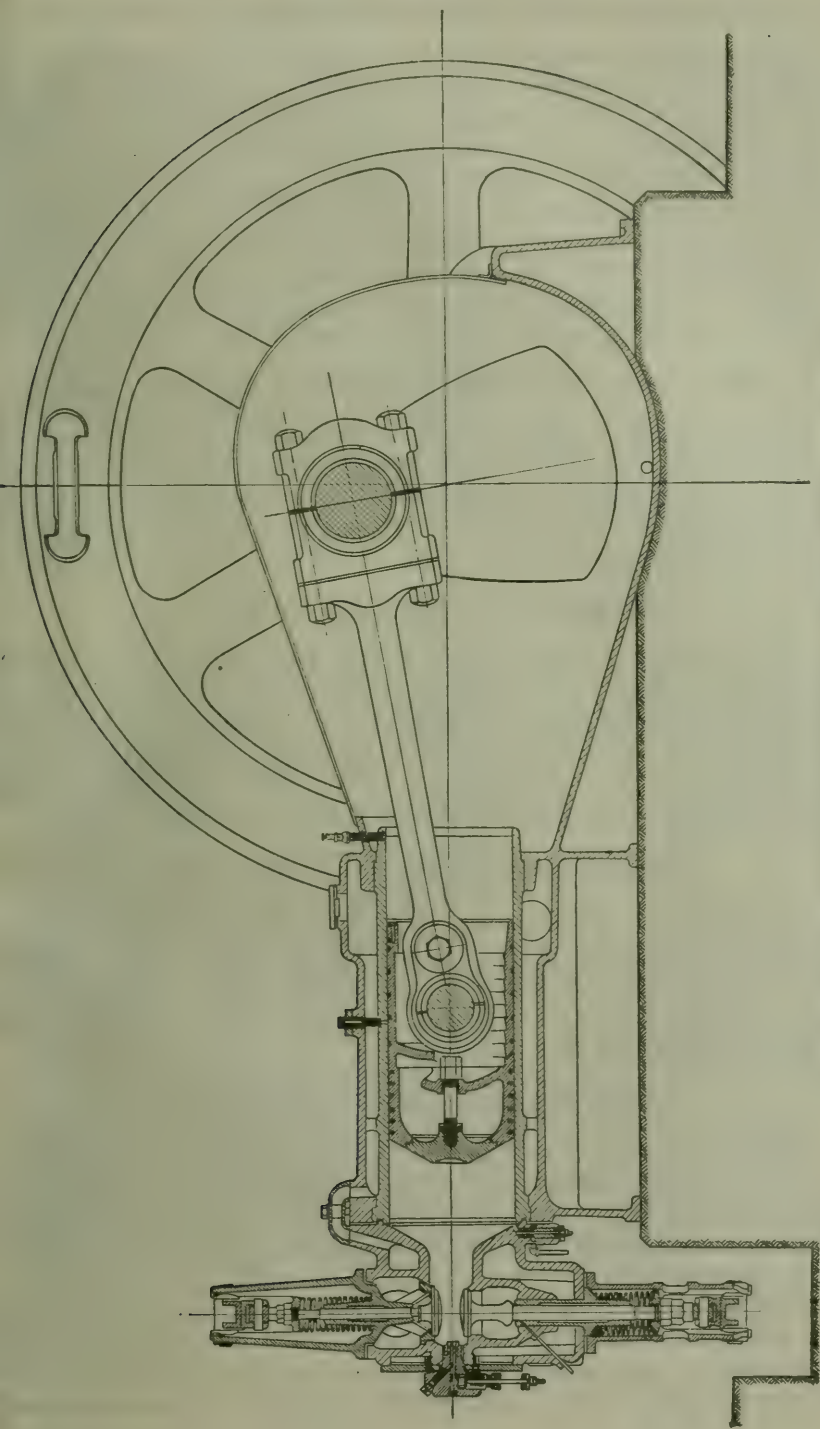


FIGURE 38.—Longitudinal section of Diesel engine built by Allis-Chalmers Manufacturing Co.

**BUSCH-SULZER BROS. DIESEL ENGINE CO., ST. LOUIS, MO.**

This company was originally the Diesel Motor Co., and later the American Diesel Engine Co., and the first to build Diesel engines in America, since 1898. It still controls a number of the earlier American Diesel patents. It is now associated with Sulzer Bros., of Winterthur, Switzerland. Types of stationary engines placed on the American market by this firm are shown in Plate VIII and in section in figure 39. They are four-cylinder units with closed crank case and four-stroke cycle, the vertical air compressor having the appearance of a fifth cylinder. The first two stages are obtained by a differential and the last by a superimposed piston.

**DOW PUMP AND DIESEL ENGINE CO., ALAMEDA, CAL.**

This company builds a vertical multicylinder, four-cycle engine, under license from Willans & Robinson, Rugby, England. It is of A-frame construction and conforms to established European Diesel engine practice. The compressor is of the direct-connected, completely inclosed Reavel type.

**FULTON IRON WORKS, ST. LOUIS, MO.**

This company builds the well-known Tosi engine under license from Franco Tosi, of Milan, Italy. The engine is built as vertical multicylinder units with four-stroke and two-stroke cycle, and with A frames and vertical two-stage and three-stage compressors, depending on the size of the engine. With few modifications, it is built along the lines developed by its European builders.

**FULTON MANUFACTURING CO., ERIE, PA.**

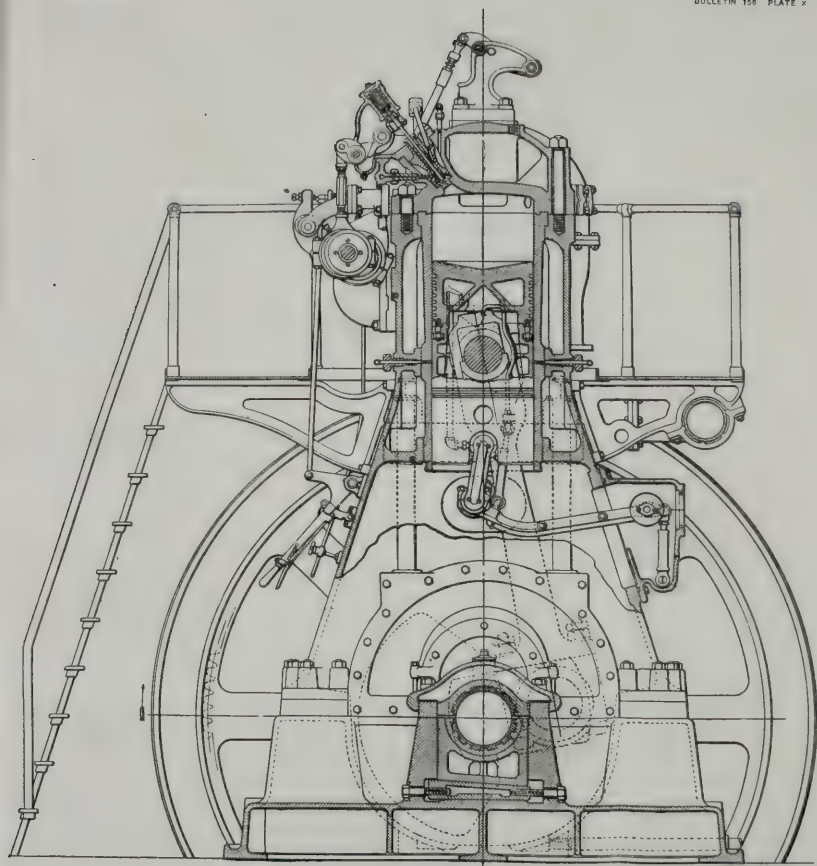
The engine of this company is of the marine type, but can be adapted to stationary work. It is built as a multicylinder engine with four-stroke cycle and with vertical two-stage compressor. It is not directly reversible, but uses a mechanical reversing gear. It is shown in Plate IX, A.

**LYONS ATLAS CO., INDIANAPOLIS, IND.**

This company developed its own engine, built as two, three, four, and six-cylinder vertical units with four-stroke cycle, all with one standard-sized cylinder of 150 horsepower. The general arrangement of this engine is shown in Plate X. Its general appearance is shown in Plate IX, B.

Admission and exhaust valves are operated through eccentrics, actuating wiper cams, and valve levers. The fuel valve is placed in the side of the cylinder head, and is also driven by an eccentric. As the fuel pump shown in figure 34 is controlled by patents owned





CROSS SECTION OF DIESEL ENGINE BUILT BY LYONS-ATLAS CO.

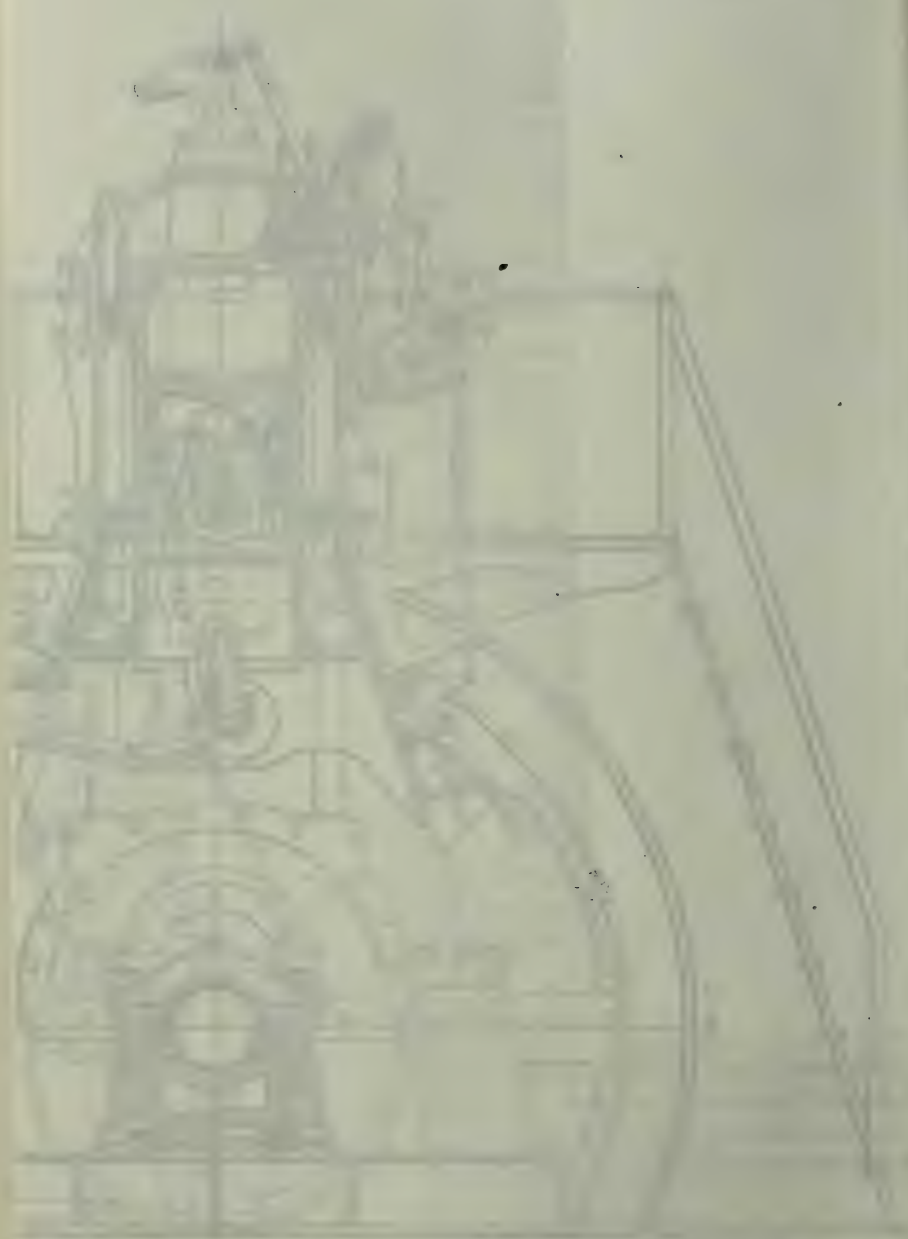
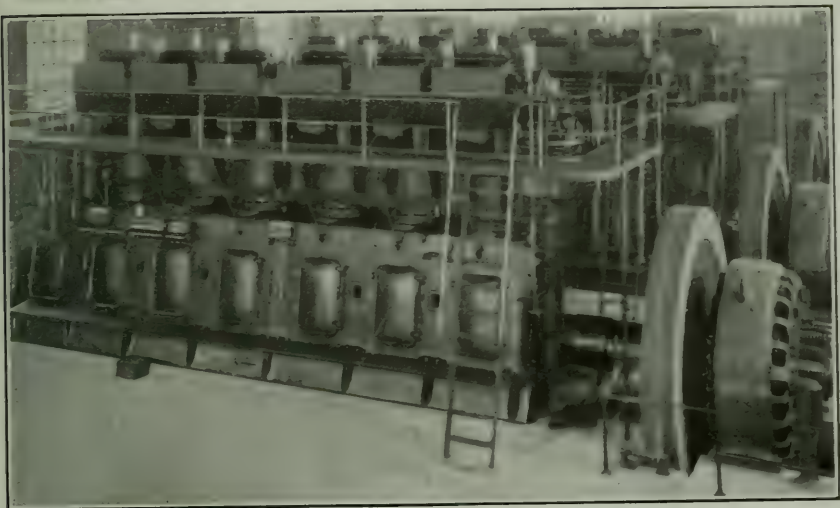
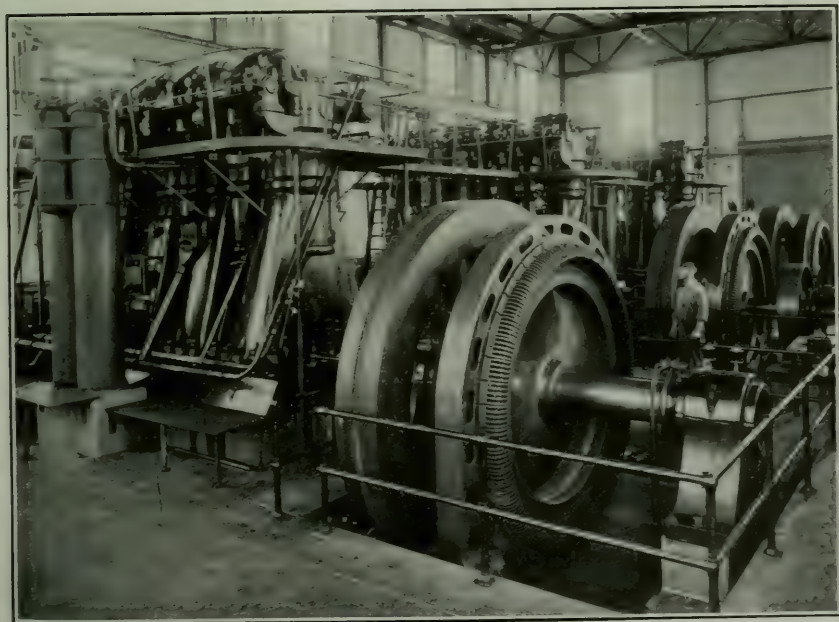


FIG. 1. THE GREAT HALL OF THE ANCESTORS, PEKING

Source: *Journal of the Royal Asiatic Society*, 1907, vol. 37, p. 100. Reproduced by permission of the Royal Asiatic Society.



A. THREE McINTOSH & SEYMOUR 1,000-BRAKE-HORSEPOWER DIESEL ENGINES.



B. THREE McINTOSH & SEYMOUR 500-BRAKE-HORSEPOWER DIESEL ENGINES





by the Busch-Sulzer Bros. Diesel Engine Co., this company designed its own fuel pump. The pump consists of a measuring plunger, whose travel is influenced by the governor. It delivers a measured volume of fuel to the forcing plunger while this is on its downward or suction stroke. Combination check and delivery valves control the flow of oil from one plunger to the other and to the fuel valve. There is a separate set of plungers for each fuel valve (per cylinder), housed in one pump chamber. Injection and starting air is furnished by a separately driven Ingersoll-Rand air compressor.

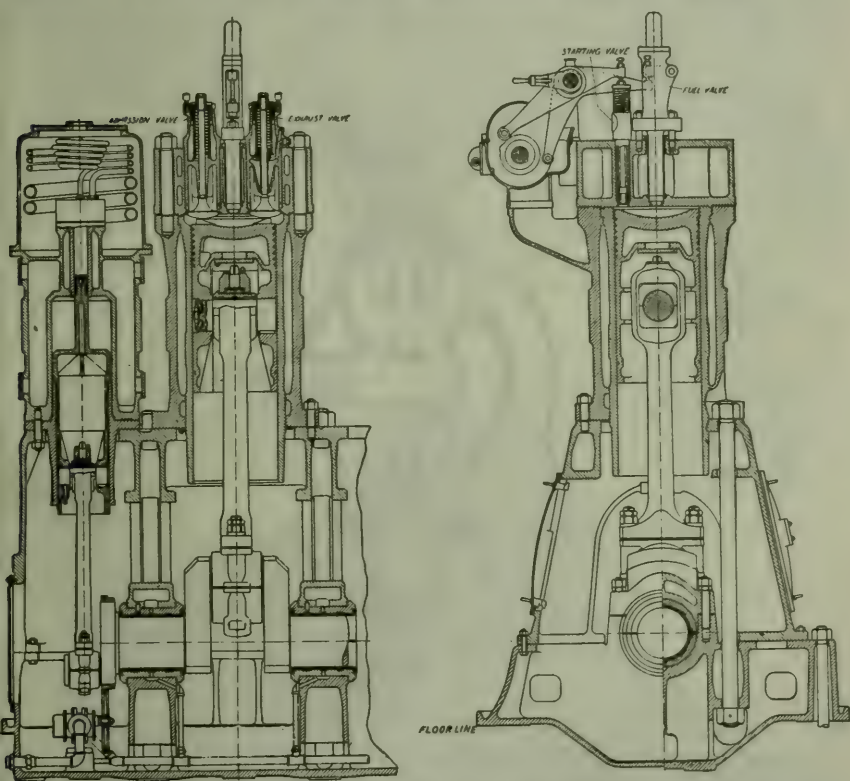


FIGURE 39.—Two sectional views of Diesel engine built by Busch-Sulzer Bros. Diesel Engine Co.

#### McINTOSH & SEYMOUR CORPORATION, AUBURN, N. Y.

This company is the American licensee of the Swedish Diesel Engine Co. (Aktiebolaget Diesels Motorer, Stockholm, Sweden) and owns the American patents and rights of that company. It builds stationary and marine engines with individual A frames and closed crank cases. They have a four-stroke cycle, and are equipped with two-stage compressors. The company states that its newer engines are fitted with three-stage compressors. The fuel valve developed by Hesselmann is used. In design the engine follows the well-established

lished practice of its European builders. Plants using the engine equipped with A frames and with closed crank case are shown in Plate XI.

**NATIONAL TRANSIT PUMP & MACHINE CO., OIL CITY, PA.**

The engine of this company is horizontal and is fitted with gas-engine head and rocker and wiper type of valve gear driven by eccentrics from the lay shaft. An open fuel nozzle is used, the injection air valve being operated from the lay shaft by means of a cam. The fuel pump and the governor are mounted together on the side of the main frame. It is driven by an eccentric from the

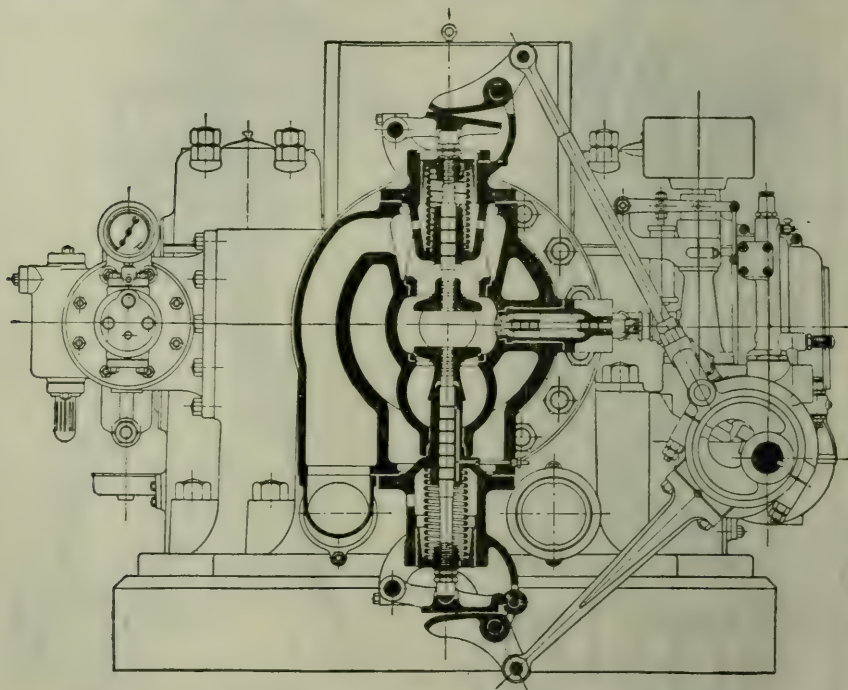
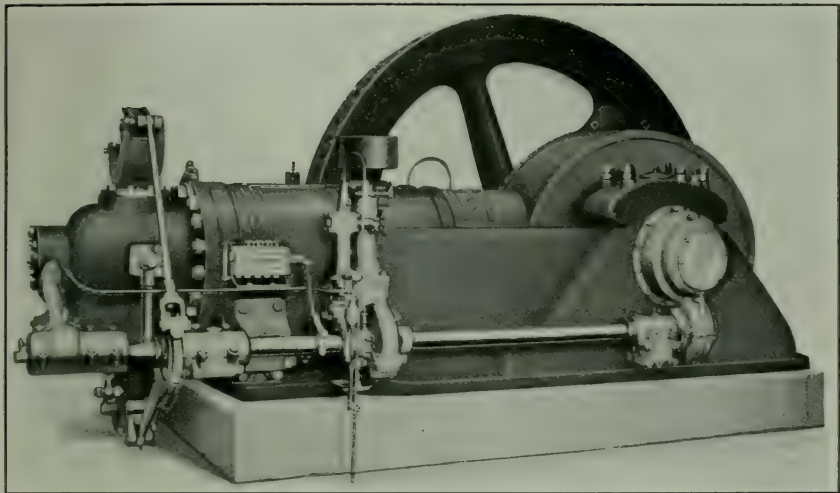


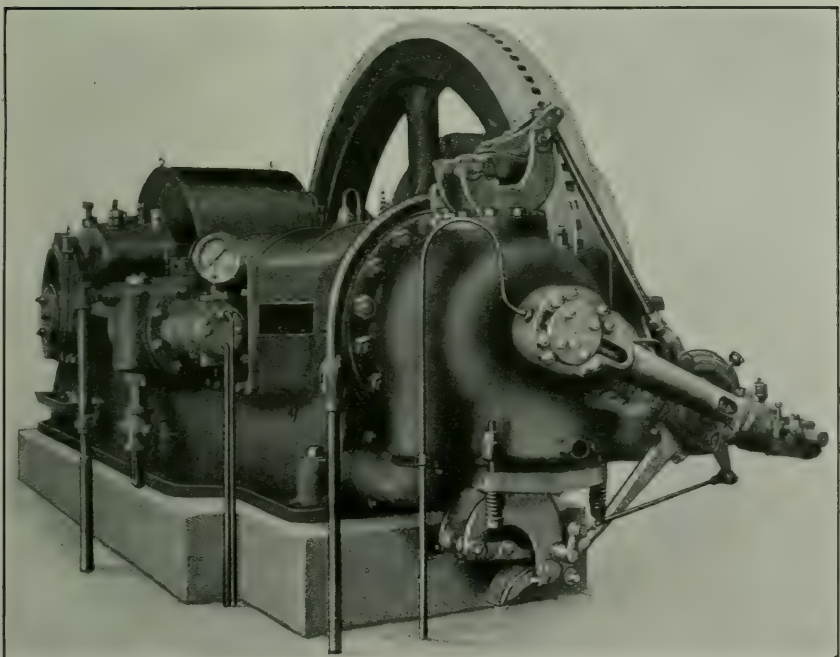
FIGURE 40.—Section through head of engine built by National Transit Pump & Machine Co

lay shaft. The upper part of the plunger is hollow and has seated on its upper end a cut-off valve which is under direct governor control. The plunger has a constant full stroke, the quantity of oil supplied being proportioned to the engine load. The supply is governed by the distance the cut-off valve is held off its seat, its action being that of a by-pass valve. The cylinder liner is removable. A two-stage air compressor, mounted on the side of the main frame, furnishes the injection air. The engines are built as single and twin cylinder units in sizes from 50 to 350 horsepower. The engine is shown in figures 40 and 41 and in Plate XII. The fuel pump is shown in figure 33 (p. 55).

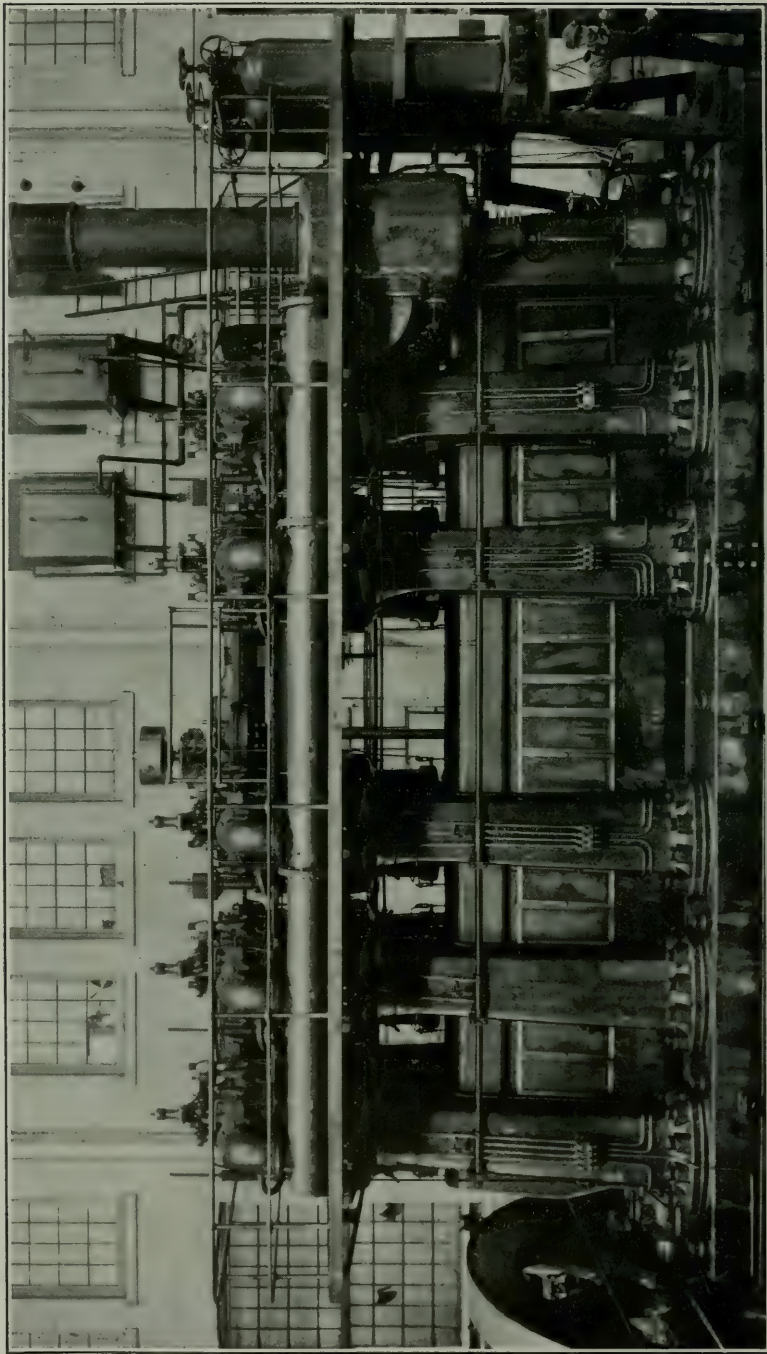




A. SIDE VIEW OF DIESEL ENGINE BUILT BY NATIONAL TRANSIT PUMP & MACHINE CO.



B. END VIEW OF DIESEL ENGINE BUILT BY NATIONAL TRANSIT PUMP & MACHINE CO.



DIESEL ENGINE BUILT BY NORDBERG MANUFACTURING CO.

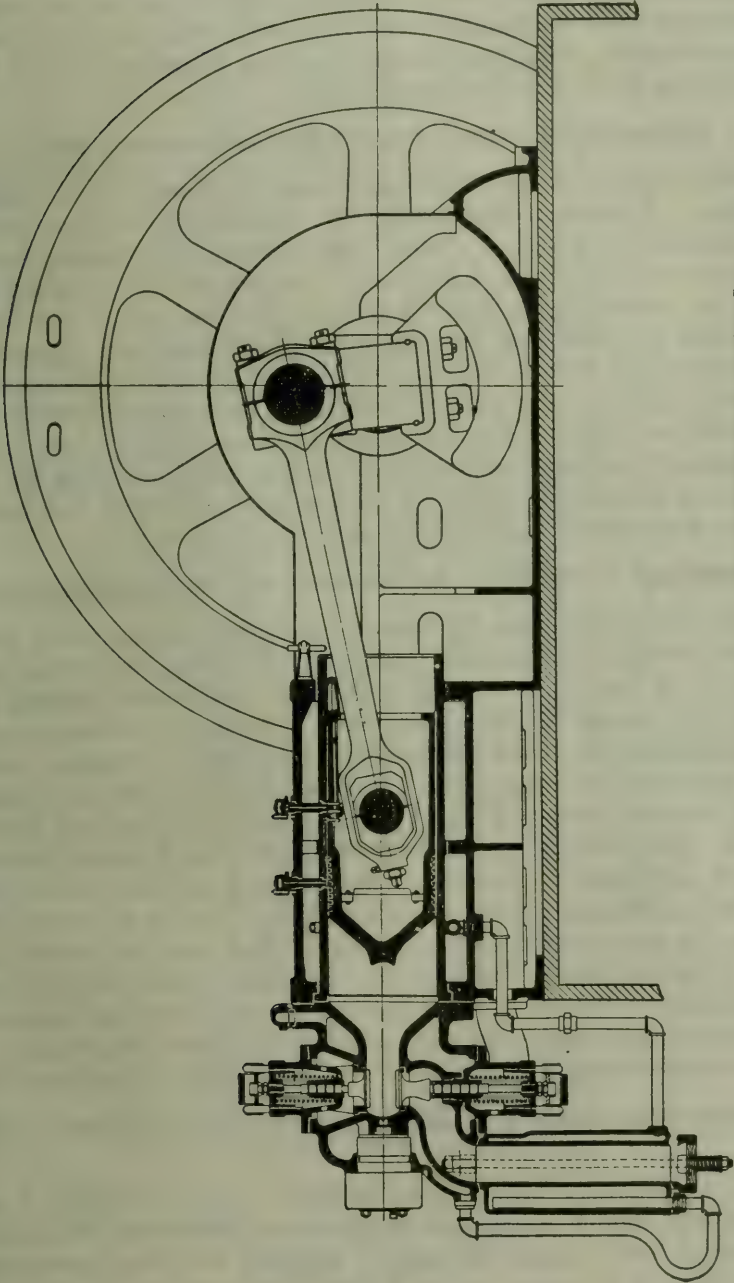


FIGURE 41.—Longitudinal section of engine built by National Transit Pump & Machine Co.



**NEW LONDON SHIP & ENGINE CO., NEW LONDON, CONN.**

This company builds both stationary and marine engines under license from the Maschinenfabrik, Augsburg, Nurnberg. The stationary engines are horizontal and are built along the lines of the well-known M. A. N. engines of this type.

**NORDBERG MANUFACTURING CO., MILWAUKEE, WIS.**

This company is the American licensee of Carels Bros., Ghent, Belgium. It builds a single-acting two-stroke engine duplicating the well-known Carels design. Plate XIII shows a five-cylinder engine of this type rated at 1,250-brake horsepower. The scavenging pump is carried by a sixth A frame. Two engines of this type furnished by Carels Bros. have been installed by the Burro Mountain Copper Co., at Tyrone, N. Mex. They are coupled to alternating-current generators and furnish current for the mines and the mills. A third unit like the above and two three-cylinder (750-brake horsepower) engines driving air compressors have been built by the Nordberg Manufacturing Co. for the same plant. A full description of this power plant has been written by Le Grand.<sup>a</sup>

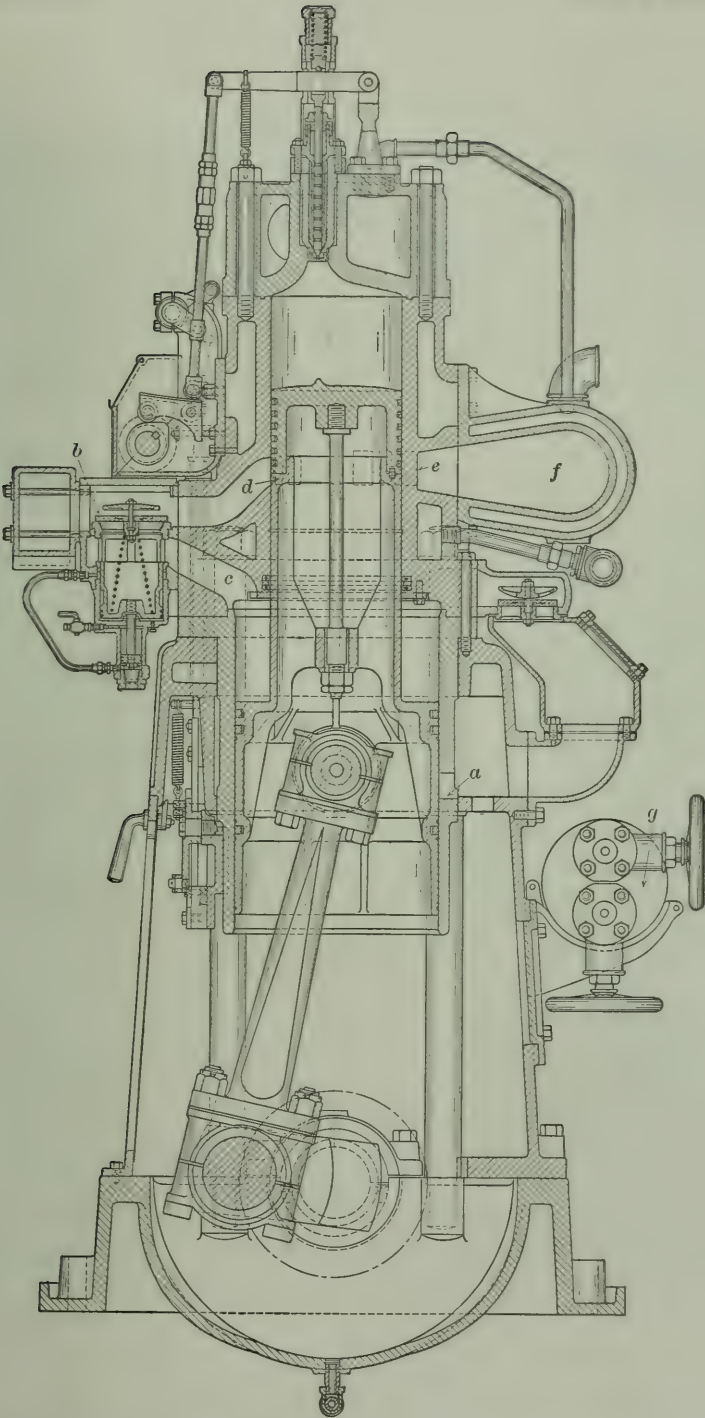
**SOUTHWARK FOUNDRY & MACHINERY CO., PHILADELPHIA, PA.**

The engine developed by this company is a single-acting two-stroke vertical multicylinder engine used chiefly for marine purposes; it is then built directly reversible. Differential pistons are used in the engine (Pl. XIV), the offset being employed to compress the scavenging air in the lower enlarged part of the cylinder, which thus forms a scavenging pump. Air is admitted into the scavenging pump through ports *a* (Pl. XIV), when these are uncovered by the piston. The scavenging air controlled by valve *b* issues through ports *c*, and when the working cylinder piston uncovers ports *d* the products of combustion are swept out of the cylinder through ports *e* into the water-cooled exhaust main *f*. Only the fuel valves are in the cylinder head. The injection air bottle is shown at *g*. Compressed air for fuel injection for starting and for reversing the engine is furnished by a three-stage compressor. The general appearance of a four-cylinder engine with the air compressor is shown in Plate XV.

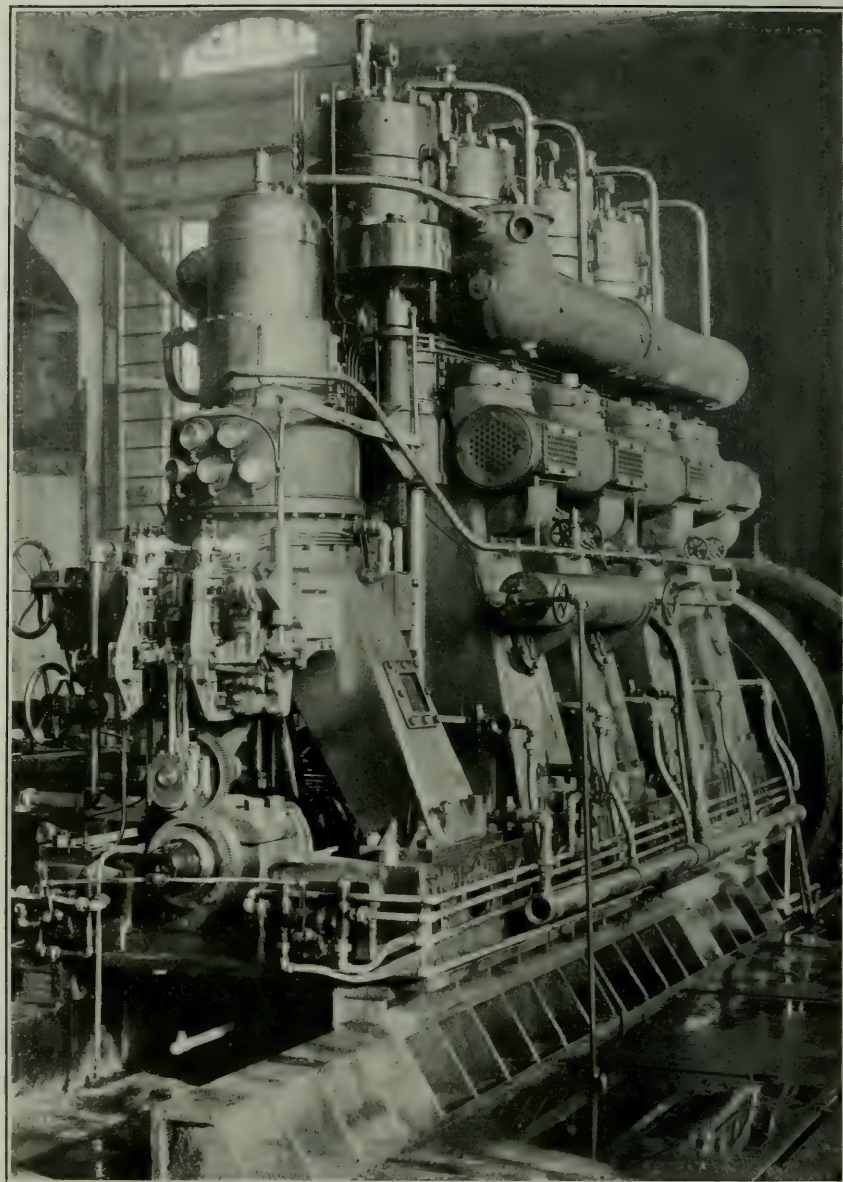
**WORTHINGTON PUMP & MACHINERY CORPORATION, NEW YORK, N. Y.**

The Snow Steam Pump Works, Buffalo, N. Y., subsidiary of the corporation, builds horizontal, single-acting Diesel engines having two-stroke and four-stroke cycles. Engines with the four-stroke cycle have all valves in the head (front) driven from a cam shaft,

<sup>a</sup> Le Grand, Charles, Power plant of the Burro Mountain Copper Co.: Trans. Am. Inst. Min. Eng., vol. 55, September, 1916, pp. 208-217.



VERTICAL SECTION OF DIESEL ENGINE BUILT BY SOUTHWARK FOUNDRY & MACHINERY CO.



DIESEL ENGINE BUILT BY SOUTHWARK FOUNDRY & MACHINERY CO.



which is engaged through miter gears by the governor shaft. The pistons are connected with a crosshead carried in a guide. The compressor is horizontal, is carried on the side of the engine frame, and is driven direct from the main shaft. It uses an open fuel nozzle and variable-stroke plunger pump, which delivers the fuel to the fuel valve during the suction stroke.

## LIQUID FUELS FOR DIESEL ENGINES.

### CLASSIFICATION OF FUELS.

Liquid fuels for Diesel engines may be divided into three groups, in which the fuels are radically different in structure, chemical constitution, and properties, and accordingly behave differently in the Diesel engine. The first group comprises fuels composed of compounds belonging chiefly to the aliphatic series, namely, a mixture of saturated and unsaturated hydrocarbons, relatively rich in hydrogen, represented by the subgroups  $C_nH_{2n+2}$ ,  $C_nH_{2n}$ , and  $C_nH_{2n-2}$  (paraffin or methane, olefiant, and acetylene series). It comprises (1) petroleum and (2) lignite-tar oils. The second group of fuels is composed principally of aromatic hydrocarbons, or benzol derivatives. It comprises (1) coal-tar oils, (2) coal tars, and (3) miscellaneous tars. The third group comprises vegetable oils, which are glycerides of fatty acids.

### GENERAL CHARACTERISTICS OF DIFFERENT FUELS.

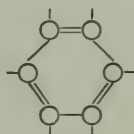
Fuels of the first group are particularly valuable for use in Diesel engines, as they lend themselves to the formation of oil gas at temperatures of  $400^\circ$  to  $500^\circ$  C. The gas is more readily formed and its ignition takes place more readily if it has a high hydrogen content, that is, a proportionately large content of saturated hydrocarbons.

The greater ease with which the oil gas is formed, ignition initiated, and the residual fuel consumed is explained by the chemical structure of these fuels. This structure is best described as being a chain, in which the atoms are held loosely together, and the more inflammable parts, like H and  $CH_4$ , are readily ignited when injected into the heated air of the engine cylinder; with the resulting rise of temperature the required high velocity of flame propagation is obtained as well as complete combustion of the residual fuel particles that do not burn so readily. Carbon particles burn the least readily. Of these fuels the saturated hydrocarbons (paraffin or methane series,  $C_nH_{2n+2}$ ) have the simplest structure, represented by  $\text{---}\bigcirc\text{---}$  methane,

$CH_4$ ;  $\text{---}\bigcirc\text{---}\bigcirc\text{---}$  ethane,  $C_2H_6$ ;  $\text{---}\bigcirc\text{---}\bigcirc\text{---}\bigcirc\text{---}$  propane,  $C_3H_8$ , etc.

The liquid fuels of the second group mentioned, composed chiefly of aromatic hydrocarbons, are low in hydrogen, are ignited with difficulty, and do not burn readily in Diesel engines. This

characteristic may be explained by their "ring" structure, which holds the atoms together in a closed ring or in combined ring groups of great stability. The ring can be broken or split only by great heat; the oxygen can not attack the "inner" ring of carbon atoms and break it, before the "outer" hydrocarbon radicals, relatively rich in hydrogen, have been ignited and burned and have furnished the high heat required to burn the carbon atoms. The outer hydrocarbon radicals themselves are not readily ignited and split from the inner carbon atoms, as they hold these tenaciously. This behavior is so pronounced that benzol ( $C_6H_6$ ), which has the simplest structure of the hydrocarbons (which can all be derived from the "benzol ring"



and are also called benzol derivatives), can not be burned in the Diesel engine, and resists combustion far more than the much heavier anthracene and creosote oils, whereas tar oils and tar rich in benzol homologues like toluol,  $C_6H_5(CH_3)$ , and xylol,  $C_6H_4(CH_3)_2$  burn more readily. An explanation for this behavior may be the attack and ignition first of the  $CH_3$  radicals, which, being richest in hydrogen, burn more readily and furnish the requisite heat for burning the rest of the hydrocarbons with high carbon content.

Petroleums rich in saturated hydrocarbons (paraffin base oils) and having the highest hydrogen content burn most readily; with a decreasing hydrogen content, an increasing resistance to rapid oxidation (combustion) is noted. Asphaltic oils and those composed chiefly of naphthenes (heavy California oils) do not burn as readily as the former type.

A product typical of the transition involved in making aromatic hydrocarbons from lignite tar oils, composed chiefly of hydrocarbons belonging to the aliphatic series, is the tar oil made from Scotch bog-head shales and cannel coals. Although many of these liquid fuels<sup>a</sup> have a high residual content (pitch) of about 50 per cent left at  $400^\circ C.$ , they burn well in the Diesel engine and leave no residue. These tars are the product of a slow distillation at low temperature and differ in physical and chemical characteristics from ordinary coal tars. According to Lunge and Köhler<sup>a</sup> they contain, besides paraffins, toluol and naphthalene, but contain little benzol and anthracene. Lacking the most valuable constituent of coal tars, benzol, they have therefore small value as a coal tar, but are valuable as a fuel for Diesel engines.

<sup>a</sup> Lunge, Georg, and Köhler, Hippolyt, *Die Industrie des Steinkohlenteers und des Ammoniaks*, Braunschweig, 1912, Bd. 1 and 2.

The resistance to ignition and burning increases still further with the coal-tar oils and is greatest with coal tars, which, with certain exceptions, are not suitable for Diesel engine fuels.

### ALIPHATIC HYDROCARBONS.

#### PETROLEUMS.

Petroleum and certain of its products are the most valuable and abundant source of fuel for Diesel engines, and the United States is fortunate in possessing large petroleum resources; the world's production of petroleum is in round figures about 60 million short tons, of which the United States produces about two-thirds.

#### DESIRABLE PROPERTIES OF PETROLEUM FUEL.

A desirable petroleum fuel for Diesel engines should have the following properties:

1. It should burn completely without leaving any residual matter in the cylinder, either in the form of soot, coke, or ash.
2. It should be free from mechanical impurities which might clog the fuel pipes, the valves of the fuel pump, and the fine fuel passages in the fuel-injection valves and nozzles, or might cause excessive cylinder wear.
3. It should be sufficiently fluid at ordinary temperatures to flow readily to the fuel pump and thence to the fuel-injection valve.
4. It should be free from water, as water lowers the heating value of the oil and may prevent its ignition.
5. It should be free from highly volatile oils, which will evaporate at ordinary temperatures and form an inflammable mixture with the air, thus introducing a fire hazard.
6. It should have a high heating value.

As is shown later, complete burning of an oil so that no combustible or carbonaceous residues remain in the cylinder, the exhaust being colorless, depends almost entirely on the type and the size of the engine, the fuel-valve and the atomizer construction, and the correct setting of the valves.

#### TESTS FOR SUITABILITY.

To determine the suitability of petroleum products as fuels for Diesel engines, they should be classified by tests as to the following properties, stated in the order of their importance:

1. Boiling-point range.
2. Ash content.
3. Mechanical impurities.
4. Heating value.
5. Water content.
6. Coke content.



7. Asphalt content.
8. Paraffin content.
9. Sulphur content.
10. Acidity.
11. Elementary composition.
12. Viscosity.
13. Flash point.
14. Burning point.
15. Specific gravity.

#### FRACTIONATION OR BOILING-POINT TESTS.

Fractionation or boiling-point tests are the most valuable, as they indicate the degree to which an oil can be vaporized, the volume of oil gas formed, and the combustibility of the oil.

The distillation is conducted at atmospheric pressure between the following temperature ranges:  $0^{\circ}$  to  $150^{\circ}$  C.;  $150^{\circ}$  to  $200^{\circ}$  C.;  $200^{\circ}$  to  $250^{\circ}$  C.;  $250^{\circ}$  to  $300^{\circ}$  C.;  $300^{\circ}$  to  $350^{\circ}$  C.; and  $350^{\circ}$  to  $400^{\circ}$  C.

As a general rule the smaller the proportion of residue remaining at a temperature higher than  $400^{\circ}$  C., and the greater the volume of vapor coming over between  $200^{\circ}$  and  $400^{\circ}$  C., the better is the oil suited for use in Diesel engines. A further valuable criterion of the burning qualities of oils that leave residues or "oil tars" at temperatures higher than  $400^{\circ}$  C. is the quantity of coke left on distilling the residue at temperatures higher than  $400^{\circ}$  C. The greater the quantity of coke the less suitable is the fuel. Fuel oils with a coke content can not be used in all engines, and a content of 5 per cent may be taken as the upper limit for all engines.

It has been noted that some crude oils that have a high proportion of constituents that produce coke and asphalt and that yield an unusually large proportion of residues at temperatures higher than  $400^{\circ}$  C. decompose (crack) at temperatures between  $300^{\circ}$  C. and  $400^{\circ}$  C. This feature is particularly noticeable in certain California oils and Mexican oils. Such an oil is unsatisfactory for Diesel engines, as only a comparatively small part of the oil is readily burned, a large part being changed to coke; the coke particles contaminate the film of lubricating oil, fill the space between the piston rings, and cover the piston, making lubrication ineffectual, causing increased cylinder wear, and requiring frequent cleaning of the engine.

The distillation test is an index of the fire hazard of an oil. If the volume of vapors passing over at  $150^{\circ}$  C. ( $0^{\circ}$  to  $150^{\circ}$  C.) is relatively large, the presence of volatile oils is indicated. It is then well to make another range test between  $0^{\circ}$  and  $50^{\circ}$  C. to determine the percentage of highly volatile oils, which affords a better indication of the fire hazard of an oil than even its flash point.

## EFFECTS OF ASH CONTENT.

Ash is the most detrimental remnant of the burning of fuel oils for Diesel engines, as it causes excessive wear of cylinders and exhaust valves. The ash is usually composed of mineral particles of great hardness, such as quartz and silicates, or oxides of iron and aluminum, which, becoming mixed with the film of lubricating oil, adhere to the piston and the cylinder walls, accumulate, and cause excessive wear. An ash content in excess of 0.05 per cent will render an otherwise excellent fuel unsuitable for use in a Diesel engine.

## MECHANICAL IMPURITIES.

The oil should be free from all mechanically held impurities of an organic or an inorganic nature. When present they should be carefully removed by filtering. Clean oils are sometimes contaminated with impurities when loaded into tank cars in which impurities or dirt have been allowed to accumulate. Impurities also may be introduced by drawing the oil from near the bottom of oil-storage reservoirs or tanks of refineries in which any sediment in the oil may have accumulated at the bottom for a long period. Likewise, at the power plant, the oil should never be drawn from too near the bottom of the main oil-storage tank, in order to prevent mechanically held bodies from being drawn into the fuel pipes and causing interruptions in the power service. Representative samples for testing are therefore important, as samples taken from near the top of a tank car or storage tank may not disclose the presence of impurities, and may show in all other respects a normal composition that is satisfactory as regards the use of the oil for fuel.

When oils are exposed to the atmosphere, complex oxidation products are formed; these, if present in small proportion only, will act like other impurities in obstructing the fine fuel passages of the fuel valves, will deposit gummy incrustations around the fuel needle, and will interfere with the satisfactory operation of the engine. The probable formation of these products should be kept in mind in storing the oil.

It is always best to provide a power installation with two small fuel filters of metal gauze in addition to the regular fuel-supply tank in the engine room, to prevent impurities from getting into the smaller fuel pipes ahead of the fuel pump or clogging the fuel nozzle. These filters should be in addition to the filters with which every good fuel pump should be equipped.

## HEATING VALUE.

The heating value of an oil is important as showing the energy available for power generation. An oil otherwise desirable is the

more valuable the greater its heating value. This should be not less than 10,000 calories per kilogram, or 18,000 British thermal units per pound. These figures refer to the mean lower heating value; that is, the heat of condensation of the water resulting from the combustion of the hydrogen in the oil should be deducted from the upper or total calorific value, or heat of combustion. In comparing the heating values of different oils, they should be reduced to a common basis or standard of the water-free and ash-free substance.

#### WATER CONTENT.

A distinction should be made between water that is so thoroughly mixed with the oil as to form an emulsion and therefore does not separate from the oil on standing, and water that does so separate. Oils containing water of the former kind will not necessarily cause a failure of ignition, as free water is not present in sufficient proportion to prevent combustion. A higher temperature is, however, required to ignite and burn such oils, and the contained water lowers the temperature of combustion. The water of such emulsions often holds mineral salts (sodium, calcium, and magnesium chlorides and carbonates, etc.) in solution, which are crystallized out with the evaporation of the water, resulting in incrustations and wear of engine parts.

Mechanically entrained water, which will separate from the oil and accumulate in the bottom of the fuel tank, will cause failure of ignition, and if it displaces the oil in the fuel valves long enough, the engine will stop.

Water lowers the heating value of the oil, as its evaporation consumes fuel; it also lowers the temperature of the combustion space. A purchaser should not pay for water and the cost of transporting it when he is buying oil. Fuel oils should not carry more than 0.5 per cent of water, and a greater water content should be subject to a penalty.

#### COKE RESIDUE.

The action of coke residue has already been discussed. Oils leaving coke on distilling can not be burned in all Diesel engines. A 5 per cent coke content is probably the upper limit. Oils with an even higher content can be burned, especially in large engines, but such oils will more quickly foul the valves and the engine cylinder, and will cause increased wear. The selection of an oil will always resolve itself into the striking of a balance between the relative first cost of a cheaper oil and the relative cost of engine upkeep. There is in the United States such an abundance of fuels suitable for Diesel engines that there is no need to have recourse to fuels that produce an unduly high yield of coke.



## ASPHALT CONTENT.

Asphalt is here termed matter insoluble in ethyl ether and ethyl alcohol (Holde's method). Most oils high in residues produced at a temperature higher than  $400^{\circ}\text{C}$ . are high in asphaltic substances. High asphaltum content makes an oil objectionable for use in engines with fuel valves that are closed by fuel needles. It tends to gum the needle and causes it to stick. By heating the oil sufficiently to make it more liquid, this objection is greatly removed. Such oils are as a rule highly viscous, and they have to be heated to cause the necessary fluidity.

High asphalt content points to a high content of constituents that will produce a coke residue. If the coke residue of an oil is satisfactory, its asphalt content may be disregarded as being of no importance.

## PARAFFIN CONTENT.

The paraffin content of an oil is important merely in its physical effect on the oil. Oils with an appreciable paraffin content may solidify or become highly viscous at low temperatures ( $0^{\circ}$  to  $-15^{\circ}\text{C}$ .). Moderate heating of oils will obviate any difficulty from the sluggishness at low temperatures due to paraffin content.

## SULPHUR CONTENT.

Sulphur in oil burned in the engine cylinder is converted to sulphur dioxide and may be oxidized to the trioxide. At the high temperature prevailing the percentage of sulphur usually present in fuels has no appreciable effect on cast iron. Even so high a sulphur content as 5 per cent in the fuel corresponds to less than 0.1 per cent by volume of sulphur dioxide in the gaseous products of combustion, which are replaced by a fresh volume of air every revolution in a two-stroke engine, or every other revolution in a four-stroke engine.

Mexican oils containing sulphur up to 5 per cent are successfully burned in Diesel engines; the exhaust pipes, however, should be of cast iron rather than steel, and the cooling of the gases should not be carried so far that the water vapor resulting from the combustion of the hydrogen will condense, as then the corrosive action of the sulphurous and sulphuric acids would be more destructive.

The corrosive action of sulphur dioxide gave considerable trouble in American Diesel engines of an obsolete type, in which an emulsion of lubricating oil and water was used in a closed crank case for lubricating the crank pins and the main bearings. Any of the gas that leaked past the piston was partly absorbed by the water in the crank case; the absorption was cumulative to the point of saturation and the sulphur dioxide badly corroded the crank shaft and the connecting rods.

Sulphur dioxide or trioxide may have a deleterious action on the cylinder oil, causing the formation of gummy substances and destroying the lubricating properties of the oil. The character of this action is still obscure, but usually an inferior lubricant or one that is adulterated or contains vegetable oils is indicated. Only pure mineral oils should be used for cylinder lubrication.

#### ACIDITY.

The fuel oils should be free from mineral acids.

#### ELEMENTARY COMPOSITION.

Knowledge of the elementary composition of a fuel oil is valuable as it discloses the carbon and hydrogen content of the oil. By stating the hydrogen content as so many parts of H available for every 1,000 parts of C, all oils are reduced to a common standard of comparison. Generally, a high hydrogen content points to the formation of a large proportion of oil gas that is easily ignited and burned.

#### VISCOSITY.

As the viscosity of an oil is a relative physical property, which changes with heat, the viscosity of fuel oils should be tested at different temperatures in the Engler viscosimeter, namely, at 20° C., 35° C., 50° C., and 75° C. Oils suitable for Diesel engines should have a viscosity of not more than 4° Engler at 75° C. If the viscosity exceeds 2.5° Engler at 20° C., the oil must be heated. Oils otherwise excellent may be too viscous at ordinary temperatures.

#### FLASH POINT.

The flash point of an oil is of value only as indication of the fire hazard. The flash point should be not below 60° C., and for oils used in engines with open nozzles, not below 70° C., as determined with an Abel-Pensky or a Pensky-Martens tester, corrected to a barometric pressure of 760 mm. of mercury. As flash-point determinations made in the open cup usually show results several degrees higher than those made in a closed tester, it is well to denote the instrument used in determination, as 60° CPM (60° C. in a Pensky-Martens tester).

#### BURNING POINT.

The burning point of an oil is the temperature at which it ignites and continues to burn in an open cup. The burning point is no criterion of the usefulness of an oil for use in a Diesel engine, save that it is a further index of the fire hazard. The nearer the burning point to the flash point, if the flash point is low, the greater is the fire hazard. A low flash point and a high burning point indicate the presence of highly volatile oils mixed with heavy oils. The burning point is 10°

to 50° C. and rarely 100° C., higher than the flash point. Extreme differences usually indicate crude oils that have not been "topped," or mixtures of volatile and of heavy residual oils. If the flash point is sufficiently high to preclude fire hazard, determination of the burning point is superfluous.

#### SPECIFIC GRAVITY OR DENSITY.

A density determination merely denotes in a general way whether a fuel oil is a residuum, a crude oil, or a distillate, but is otherwise of no significance in indicating the value of a fuel oil for use in Diesel engines. High viscosity is often mistaken for high density, although as regards Mexican and Californian crude oils, a dense oil is as a rule sluggish. Density is of importance when freight rates are based on volume measurement instead of absolute weight. Density values should be referred to a standard temperature of 15° C.

Petroleum fuels as herein classified comprise heavy crude oils that do not contain sufficient volatile oils to warrant the expense of "topping;" "topped" oils, from which the benzene has been removed; "gas oils," or oils distilled at relatively high temperature, the residues of which are used for road oils or converted into asphaltum; and residuums, generally classed as fuel oils, which result from oil-refining operations.

#### ENGINE FACTORS AFFECTING COMBUSTION.

Manufacturers of Diesel engines have constantly striven to improve and construct the engines with a view to burning a greater variety of fuels, and in particular to make possible the burning of the heavier fuel oils and residues that are produced in great abundance in oil refining.

The following factors influence the thoroughness of the combustion: Size and speed of engine; atomizer construction and atomization of the fuel; degree to which the fuel spray is mixed with the air of combustion; and the shape of the combustion space.

In small engines the provision of water-cooled surfaces for the cylinder and the cylinder head relatively large in proportion to the size of the combustion space is more effective than a similar provision in larger engines. Effective cooling is especially noticeable at low engine loads. The engine speed affects the speed of the fuel injection. In high-speed engines the fuel is injected during a period of one one-hundredths of a second, whereas in large low-speed engines the injection may last over a period of one-twentieth of a second. There are definite limits to the velocity with which a fuel can be burned; aside from being influenced by the character of the fuel, the velocity is chiefly controlled by the speed of the chemical reaction. As a part of the fuel burns with the formation of carbon dioxide and



water vapor, and possibly carbon monoxide, these gases reduce the activity of the oxygen toward the remainder of the fuel. With more time available for burning a fuel, and with equally thorough atomization, the combustion will be more thorough or, what is more important, the higher may be the boiling point of the oil.

#### ATOMIZER CONSTRUCTION.

The importance of thorough atomization has already been discussed. Heavy oils and residues demand higher injection-air pressure, so that the fuel will be broken into extremely small particles. The boiling-point temperatures of oils advance with an increase in pressure; further, the higher the boiling points of the oils the smaller is their vapor volume and the greater the volume of air required for their combustion. Although gasoline vapor requires for combustion about 40 times its own volume of air at the same temperature and pressure, a heavy petroleum vapor requires about 100 times its volume of air. With the increase in the volume of air, the diffusion of the fuel vapor through the air is much reduced and the obtaining of an intimate mixture of fuel and air becomes much more difficult. To bring each molecule of fuel vapor into contact with 100 or more molecules of air needed for combustion, requires a most intimate mixture of fuel and air. If this mixture is incomplete, the more volatile oil particles, which require relatively much less air, will burn first and leave particles with high boiling point incompletely burned, causing the formation of soot and oil tar.

#### SIZE AND SHAPE OF COMBUSTION SPACE.

A small combustion space, with highly-compressed air, greatly aids the thorough mixing of oil vapor and air. These considerations also explain why the combustion space of the cylinder should not be split up or have pockets, but have smooth walls so that the fuel particles can travel through the air more readily. For the same reason, the piston floor should be so shaped and the atomizer nozzle so constructed that the fuel spray will be spread over the entire cylinder area and, in being propelled forward by the force of the injection, air will be diffused through the air of combustion. The high temperature in a Diesel engine cylinder is another factor that results in much increased speed of chemical reaction.

#### SPECIAL METHODS USED IN BURNING HEAVY OILS.

Certain important adjustments, based on the foregoing considerations, are made for burning heavy oils, as follows: Increased injection-air pressure and smaller nozzle orifice to insure more complete atomization of the fuel; higher compression in the engine itself to procure higher temperature of the air of combustion; and heating of the oil

to as much as 180° F. to make it more fluid and more easily atomized. These different adjustments require considerable experience on the part of the operating engineer, who must take into account the character of the fuel used.

For oils that contain asphaltum and much oil tar at temperatures higher than 400° C. the open-nozzle type of fuel valve is more suitable, as this valve has no fuel needle, which easily becomes fouled when there is much asphaltum or tar present. In this valve the valve needle comes in contact with the injection air only—never with the fuel, which also obviates the necessity of grinding the fuel needle, as required in the closed-nozzle type of fuel valve.

Highly viscous oils, beside having to be heated, require that the engine be started and operated with gas oil until the discharge engine-cooling water is hot enough to render the viscous oil sufficiently fluid.

#### TWO LARGE CLASSES FOR PETROLEUM FUELS.

Suitable petroleum fuels may be divided into two classes—those that can be used in all Diesel engines, and those that can be used only in specially equipped engines. In the first class can be placed all fuels with sufficiently high flash point to minimize fire hazard, contain no asphaltum nor mechanical impurities, and have a mean lower heating value of 18,000 British thermal units per pound. Fuels of the second class have to be tried first in the engine, which often has to be especially equipped to burn them successfully.

#### SPECIFICATIONS FOR FUEL OILS SUITABLE FOR DIESEL ENGINES.

The following specifications for fuel oils suitable for Diesel engines, published by the Rumanian Section of the International Petroleum Commission in 1912 (*Cahier des charges types pour les fournitures de produits Romains de pétrole*; Bucharest, Imprimerie de l'Etat, 1912), are of interest as they describe what oils of the first class should be.

1. The specific gravity (density) must be 0.860 to 0.895 at 15° C., to be determined with officially standardized aërometers. The density is to be corrected for each degree of temperature above or below the standard by  $\pm 0.0007$ .

2. The flash point measured by the Martens-Pensky tester must be not lower than 60° C.

3. Tests to determine the boiling-point range must be made with a 100-c. c. sample, according to standard methods in an Engler fractionation flask; at least 90 c. c. of the oil should be distilled at 350° C., and if distillation is continued, until the residue is coked, the coke should not weigh in excess of 0.5 gram.

4. Acidity. The heavy gas oil must be free of mineral acids. For testing, the use of methyl orange as applied to kerosene (lamp oil) should be used.

5. Water and mechanical impurities. The oil should be entirely clear and should not contain any suspended matter visible to the naked eye after filtering.

6. Ash content. On evaporating 50 grams of the oil to dryness in a platinum dish, no weighable quantities of ash must remain.

7. The viscosity must not exceed 2.50° Engler, measured with the Engler or the Engler-Ubbelohde viscosimeter according to standard methods.

8. Heating value, determined with the Berthelot-Mahler calorimeter, should be at least 10,000 calories per kilogram (18,000 British thermal units per pound).

It will be seen that these specifications do not allow a coke residue of even 0.5 per cent, and allow no ash content at all. It is evident that the life of an engine is materially increased, and that wear and tear and the cost of maintenance are decreased when oils meeting these specifications are used. Some oil refineries in the United States have put on the market gas oils that meet practically all of these requirements, and that are sold at a small advance in price over those of heavy fuel oils and residues. The fuel consumption of Diesel engines is so small that the increase in price is hardly reflected in the cost of the power, at least of small plants; in fact, the reduced wear and tear on the engine, the less frequent fouling of the valves, and the knowledge that the oil is of constant composition, eliminating readjustments of the fuel valves, is sure to offset the slight increase in cost. This advance over the cost of ordinary fuel oils varies from 15 to 25 cents per barrel of oil, depending on locality; on a basis of 0.5 pound of oil per effective brake horsepower-hour and 640 horsepower-hours per barrel of oil (at 320 pounds) the increased fuel cost amounts to 15/640 to 25/640 cent, or about 0.025 to 0.04 cent per brake horsepower-hour.

The oil specifications of a leading American builder of Diesel engines require that the oil shall have "not more than 10 per cent residue; this residue is that remaining after a sample of the oil has been reduced to approximately constant weight in a closed furnace for 120 hours at a temperature of 300° C. Gravity at 60° F. not heavier than 20° B.; not lighter than 40° B. The gravity may be heavier than 20° B., the limitation being that it shall readily flow to and be handled by the fuel pumps of the engine, to which end artificial heating may be necessary with heavy oils and in a cold climate.

"To insure the safe storage of the oil, uninterrupted operation of the engine, and minimum wear and tear, it is recommended that fuel oil also comply with the following: Flash point, between 125° and 250° F.; burning point, between 160° and 300° F.; acid, not over a trace; sulphur, not over 1.5 per cent; water, not over 0.3 per cent; ash, not over 0.01 per cent."



In marked contrast with the foregoing requirements are the fuel specifications of one of the foremost European manufacturers of Diesel engines, a large number of whose engines are operated successfully in Mexico on heavy Mexican oils. These specifications are as follows:

1. Flash point, determined by Abel-Pensky or Pensky-Martens testers, to be not lower than  $70^{\circ}\text{C}$ .
2. Viscosity not to exceed  $4^{\circ}$  Engler at  $75^{\circ}\text{C}$ .
3. Asphaltum is not objectionable as long as the viscosity of the oil does not exceed that stated.
4. The oil should be free from water, grit, loam, or similar impurities, and should contain no sand.
5. Sulphur up to  $2\frac{1}{2}$  per cent is not objectionable.

#### LIGNITE-TAR OILS.

Lignite-tar oils are the product of slow distillation at relatively low temperature of lignites and bituminous shales. They are composed chiefly of hydrocarbons of the aliphatic series, and form a valuable fuel for Diesel engines, but are chiefly of local importance in countries having lignite beds but no supply of cheap petroleum fuels.

#### AROMATIC HYDROCARBONS.

##### COAL-TAR OILS.

Coal-tar oils as used at present for Diesel engines are mixtures of the coal-tar fractions naphthalene oil and anthracene oil. From the first the naphthalene is mostly removed; and the tar oil should be as free as possible from anthracene and its homologues. These oils are mixtures of aromatic hydrocarbons. Their structural difference from the hydrocarbons of the aliphatic series has already been noted, as well as the greater difficulty of burning these oils in the Diesel engine. These oils have high self-ignition points. A cold engine can therefore not be started with them. It must first be heated by using a gas oil. Such oils will not prove satisfactory for operating an engine at fractional loads, unless an ignition oil (gas oil) is injected ahead of the tar oil to produce the heat necessary to initiate combustion; otherwise, particularly at low load, the exhaust will be very smoky.

Small engines that work with lower cylinder temperatures than those in large engines, and at higher speed, do not burn the tar oil as well as do large engines. Large engines operating continuously at full or nearly full load can burn tar oils successfully without the use of an ignition oil except for starting.

Tar oil is the principal fuel for Diesel engines in Germany, where about 1,000,000 horsepower is represented. As Germany does not

command petroleum resources of any importance and there are heavy duties imposed on petroleum fuel oils, Diesel engines were early adapted to burn the tar oils.

It is interesting to note here that the late Dr. Diesel advocated that all gas coals should be coked, so that the valuable by-products would not be wasted. The coke was then to be used to produce heat and power, furnishing a smokeless fuel. In Germany all coke is made in by-product ovens, all beehive ovens having been abolished many years ago, but Dr. Diesel wanted to see the coking extended to all coking coals mined, and to have coke instead of coal used for all metallurgical, heating, and power purposes (in locomotives, large boiler plants, and gas-producer plants). If this practice were carried into effect, Germany could develop 10,000,000 horsepower from coal-tar oils and coal tars alone, without sacrificing benzol and other valuable by-products of the coking operations; in addition there would be the possibility of producing power from the coke-oven gas, and the coke, with less waste, could be put to all the uses that bituminous coals serve.

Much of the coal made into coke in the United States is coked in beehive ovens. By the gradual change from this wasteful practice to that of coking in by-product ovens new uses for the coal tars as Diesel engine fuels could be found, without materially curtailing the production of the lighter oils, as parts of the middle and heavy oil fractions constitute the chief tar oils used in Diesel engines.

#### SPECIFICATIONS FOR TAR OILS.

Limiting values of the chemical and physical properties of typical coal-tar oils are given in Table 2 following, from Constam and Schläpfer.<sup>a</sup>

TABLE 2.—*Limiting values of physical properties of typical coal-tar oil.*

|                                                                     |                  |
|---------------------------------------------------------------------|------------------|
| Specific gravity at 15° C.....                                      | 1.006 to 1.110   |
| Flash point, °C.....                                                | 66 to 121        |
| Burning point, °C.....                                              | 84 to 160        |
| Self-ignition point in platinum crucible and oxygen stream, °C..... | b 550            |
| Boiling-point range:                                                |                  |
| Light oil, 0° to 170° C., per cent by volume.....                   | 0 to 12          |
| Middle oil, 170° to 230° C., per cent by volume....                 | 0 to 80          |
| Heavy oil, 230° to 270° C., per cent by volume...                   | 1 to 36          |
| Anthracene oil, 270° to 350° C., per cent of volume.                | 9 to 66.5        |
| Residue above 350° C. (pitch), per cent by volume.                  | 1 to 34.5        |
| Water, per cent.....                                                | 0 to 1.82        |
| Ash, per cent.....                                                  | 0 to 0.04        |
| Heating value:                                                      |                  |
| Calories per kilogram.....                                          | 8,845 to 9,125   |
| British thermal units per pound.....                                | 51,921 to 16,425 |

<sup>a</sup> Constam, E. J., and Schläpfer, P., Ueber Treiböle: Ztschr. Ver. deut. Ing., Bd. 57, 1913, p. 1621.

<sup>b</sup> Approximately.

|                                                        |                    |
|--------------------------------------------------------|--------------------|
| Heat of combustion of ash and water-free oil:          |                    |
| Calories per kilogram.....                             | 9, 236 to 9, 506   |
| British thermal units per pound.....                   | 16, 625 to 17, 110 |
| Carbon, per cent.....                                  | 87. 1 to 91. 4     |
| Hydrogen, per cent.....                                | 6 to 7. 8          |
| Oxygen and nitrogen, per cent .....                    | 1. 4 to 4. 9       |
| Sulphur, per cent.....                                 | 0. 4 to 0. 9       |
| Hydrogen available to 1,000 parts of carbon, parts.... | 62 to 83           |
| Coke, per cent.....                                    | 0. 4 to 3. 6       |
| Free carbon and mechanical impurities, per cent.....   | 0 to 0. 2          |
| Naphthalene, per cent.....                             | 0. 8 to 10. 5      |

The following specifications covering tar oil suitable for Diesel engines have been adopted by the German Tar Product Syndicate and certain manufacturers of Diesel engines:

1. The tar oil must contain not more than 0.2 per cent of solid matter insoluble in xylol; the noncombustible constituents must not exceed 0.05 per cent.
2. The water content must not exceed 1 per cent.
3. The residual coke must not exceed 3 per cent.
4. At least 60 per cent by volume of the oil must be distilled below or at 300° C.
5. The mean lower heating value must be not less than 8,800 calories per kilogram (15,800 British thermal units per pound).
6. The flash point must not be below 65° C.
7. The oil must be thoroughly fluid at 15° C. If the oil is cooled to 8° C. and allowed to stand for half an hour, no solids must separate out.

In engines using coal-tar oils, all fittings coming in contact with the oil must be made of cast iron or nickel steel, as cast steel and wrought iron are severely attacked by coal-tar oil.

#### COAL TARS.

According to the method used in coking the coal, coal tars may be classified as horizontal-retort tars, inclined-retort tars, vertical-retort tars, chamber-retort tars, and tars made in coke-oven retorts. Of these coal tars, with a few exceptions, only those recovered in the manufacture of coal gas in vertical retorts and those recovered from the production of coke in by-product coke ovens are suitable for use in Diesel engines. In exceptional instances the coal tar from inclined retorts has been used. Coal tars vary so much in composition with the different coals and with the kind of retorts and coke ovens that a given tar must be carefully tested, chemically and in the engine, before a decision can be reached as to its usefulness as a fuel for Diesel engines. They almost invariably contain mineral constituents introduced from the retort or coking chambers, and if these constituents are of a silicious character they are particularly destructive in action on the cylinders and valves.



Tars made in horizontal retorts can not be used in Diesel engines because they contain only a small proportion of light oil and a relatively large proportion of heavy and anthracene oils, and more than half of the tar consists of pitch. The coke residue is therefore high, being often in excess of 30 per cent, with a correspondingly high free-carbon content; likewise the naphthalene content is high, as the primary distillation products are decomposed further in horizontal than in vertical retorts, owing to the uniformly higher temperatures used in horizontal retorts. These tars are black and at ordinary temperatures are highly viscous. They have the highest flash, burning, and ignition points of the gas-retort tars.

Intermediate between horizontal-retort and vertical-retort coal tars are those from inclined retorts.

Vertical-retort tars and chamber-oven tars are successfully used as fuel in Diesel engines; these tars have a high hydrogen content and therefore a higher heating value than the tars made in horizontal retorts. They are relatively high in light oils; their flash and burning points are therefore low, and they are thin fluids. They have relatively little pitch at temperatures higher than  $350^{\circ}\text{C}$ ., and the free-carbon content, the coke residue (average about 6 per cent), and the naphthalene content are the lowest of all the coal tars. They are relatively low in water content and in ash.

Coke-oven tars differ from those mentioned previously in that they are "debenzolized" in the process of production, and are therefore black, thick, heavy liquids. Their specific weight, flash points, burning points, and ignition points are the highest of all the coal tars. They have high boiling points, and contain no light oil and little "middle" oil, but are relatively high in heavy and anthracene oils. They contain 50 to 65 per cent of pitch, being therefore correspondingly high in free carbon and in coke residue. The composition of these tars varies considerably, as they are produced from a great variety of coals and in different types of coke ovens.

All coal tars are relatively high in ash content, mechanical impurities, and water. Moreover, the free carbon and the coke residue foul the cylinders and the valves and these tars vary in composition and properties. Consequently they become more valuable for fuel for Diesel engines after having been subjected to a distillation or refining process. By this process the coal-tar oils already described are obtained, which are distinguished by greater purity and uniformity in composition.

In many European cities sufficient coal tar is derived from local gas works to supply the fuel requirements of the Diesel engines that provide the cities with light and power. In Switzerland particularly all power plants operated by Diesel engines, of which there is a large number, are now using gas-works coal tar or tar oils made from such tars, as the supply of petroleum fuel from the Galician fields has been cut off by the war.

## WATER-GAS AND OIL-GAS TARS.

Water-gas and oil-gas tars are waste products resulting from the decomposition of gas oils from petroleum and of tars from lignite, bituminous shales, and oil shales in the manufacture of illuminating gas. The properties of these tars vary greatly according to how highly the retorts have been heated and therefore may partake more of the properties of the initial gas oil or approach those of coal tars. Their use in Diesel engines, therefore, depends entirely on their chemical and physical properties. Tars that are high in hydrogen content and low in free carbon make excellent fuels for Diesel engines, whereas those high in free carbon and naphthalene are unsuitable.

Coal-tar oils and coal tars as Diesel-engine fuel have no immediate importance in the United States, as there is an abundance of the more desirable petroleum fuels; they may, however, be of future economic importance, especially if our coal-tar production is increased with an increase of coking operations in ovens that permit the recovery of by-products.

## VEGETABLE OILS.

## ARACHIS OIL.

Arachis or ground-nut (peanut) oil is obtained by pressing the seeds (peanuts) of *Arachis hypogaea*, a leguminous plant, growing profusely in tropical countries. The seeds are said to contain 38 to 50 per cent of oil and the pods 4 to 5 per cent. This oil can be successfully burned in Diesel engines, and it may prove a valuable source of fuel where there are no cheaper liquid fuels, as in the interior of the colonies of some foreign countries. With that object in view its use has been developed by the French Ministry of Colonies.

## PALM OIL.

Palm oil is obtained from a number of different species of palms, but chiefly from the fruit pulp of the oil palm (*Elaeis Guineensis*) growing in West Africa. The oil is a fatty substance which melts into a clear oil at about 90° F.

## COMPOSITION AND PROPERTIES OF TYPICAL LIQUID FUELS.

The composition and the properties of typical liquid fuels are shown in Plate XVI. It will be noted that with an increase in density the heating value drops. There are marked differences in the viscosities of the different fuels, necessitating the heating of those that are not refined products. There is also much variation in the temperature at which the oils evaporate and in the degree of evaporation. The Galician gas oils, which are derived principally from paraffin-base oils, are cited as an example of liquid fuel extensively used in Diesel engines abroad. Only small differences between the properties of American oils and those of Mexican oils will be noted, except that

the latter have an appreciable sulphur content. The Texas fuel oils represented in Plate XVI are excellent fuels for Diesel engines, so long as the permissible ash content is not exceeded.

The California crude oils represented are unsuitable for use in Diesel engines, chiefly because of their excessive ash content and their high coke content and their decomposition at  $300^{\circ}\text{C}$ . or at a slightly higher temperature. The oils represented in the figure are not cited as being representative of California crude oils generally, but as affording a comparison between oils suitable for use in Diesel engines and oils that are not suitable for such use. The California residuum cited meets all the requirements of a desirable fuel, the ash content not being excessive, although the boiling points range fairly high and the oil has to be heated for use.

Many Mexican crude oils, as indicated, have a low heating value on account of the large water and sulphur contents; likewise, their ash content is frequently so high as to make them worthless for fuel for Diesel engines. Many contain sufficient proportions of light oil to lower their flash points and their burning points to such a degree as to make them constitute a serious fire hazard. At temperatures higher than  $300^{\circ}\text{C}$ . they decompose, leaving large proportions of coked residue.

### LUBRICATING OILS FOR DIESEL ENGINES.

Two kinds of oils are used for the lubrication of Diesel engines—bearing oil and cylinder oil.

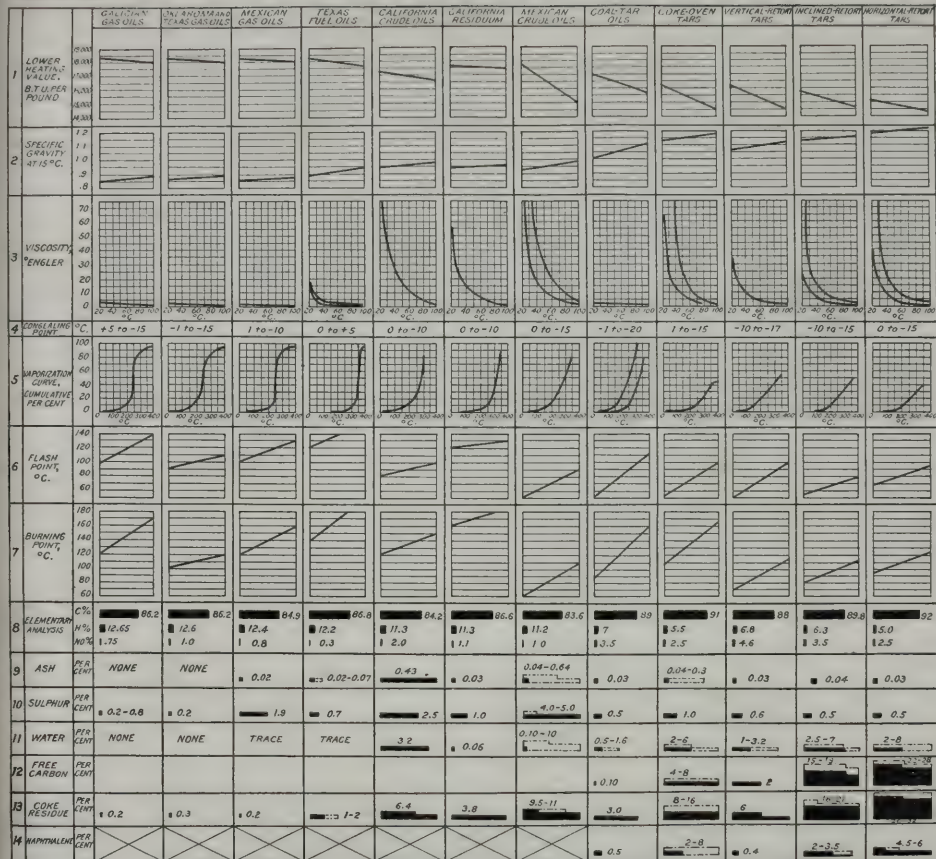
#### BEARING OIL.

Bearing oil is used for the lubrication of the main, the crank-pin, and the piston-pin bearings, and of the cam-shaft bearings, and for the oil baths into which the cams and the camshaft dip. A pure mineral oil will fill these requirements if it has a maximum viscosity of 7° Engler at  $50^{\circ}\text{C}$ ., if it does not congeal at a temperature higher than  $-10^{\circ}\text{C}$ ., and if it has a flash point of  $180^{\circ}$  to  $200^{\circ}\text{C}$ . The oil should be entirely soluble in benzene, forming a clear solution without any residue; also, it should be entirely free from acids, resinous substances, and vegetable matter.

#### CYLINDER OIL.

Cylinder oil is used for the lubrication of the working cylinders and the air-compressor cylinders. It should be stable at the relatively high temperatures of the piston and the cylinders of a Diesel engine, and should maintain the desired lubricating properties at these temperatures; it should not vaporize readily, nor should it decompose at a temperature lower than  $300^{\circ}\text{C}$ .





COMPOSITION AND PROPERTIES OF TYPICAL LIQUID FUELS.



## DESIRABLE PROPERTIES OF CYLINDER OIL.

A suitable cylinder oil should have a viscosity of  $9^{\circ}$  to  $10^{\circ}$  Engler at  $50^{\circ}$  C., and should flow at  $-5^{\circ}$  C. Its flash point should not be below  $240^{\circ}$  C. It should be a pure mineral oil free from asphaltum and pitch and resinous substances, and should be completely soluble in benzine, leaving a clear solution without any residue. A piece of sheet brass covered for 24 hours with the oil, at  $100^{\circ}$  C., should remain perfectly bright and show no action by acids, nor should it be gummed by any substance. A thin layer of the oil spread on the sheet of brass should contain no solid particles perceptible to the sense of touch.

Even the best oils will become partly oxidized in the compressor and engine cylinders and form complex pitch-like substances and insoluble compounds, which absorb fine metal particles from the compressor cylinder or dust that is carried into the cylinder with the air. With good oils it takes many months to form such accretions, and their harmful effect can be entirely avoided by a periodical inspection and cleaning of the cylinders.

Cylinder oils should not be decomposed in the working cylinders nor the air-compressor cylinders, as decomposition permits the more volatile part of the oil to burn and form carbon, which covers the piston, causes the piston rings to stick, destroys the lubricating effect of the oil, and results in the scouring of the pistons and cylinders, and sometimes in the seizure of the pistons.

Users of Diesel engines can make no more serious mistake than to purchase a low-priced cylinder oil in an attempt at economy. A high-grade cylinder oil, sparingly used, will prove most economical as regards both the quantity of oil consumed, and the cost of engine upkeep. Oil analyses alone are insufficient to govern the selection of a desirable oil. The best way of testing the suitability of an oil is to note its performance in a Diesel-engine cylinder. The best guide in the purchase of lubricating oils is the recommendation of the engine builders, as their recommendation is usually based on the results of protracted tests of different lubricating oils.

## IMPORTANT ADVANTAGES OF DIESEL ENGINES.

## LOW PRICE OF FUEL.

The high commercial value of the Diesel engine lies in its unsurpassed fuel economy and in its ability to burn low-priced liquid fuels with high boiling points. This great fuel economy is not confined merely to medium or large-sized units, for the smallest engine has nearly the same fuel economy as the largest. In this respect Diesel engines differ from steam engines, which become economical only in large sizes.



**SELF-CONTAINED PRIME MOVER.**

The Diesel engine possesses the further great advantage of being a self-contained prime mover, its only auxiliary being the air compressor, which in all high-grade engines is driven direct from the engine crank shaft, and is mounted on the engine frame. If directly connected to a direct-current generator or an alternator with excitor, it comprises with the switchboard equipment a complete electric generating station. Its space requirements are therefore small, and it can be placed in basements of buildings and hotels.

**MINIMUM FUEL-STORAGE SPACE REQUIRED.**

The small fuel requirements of the engine reduce to a minimum the the space necessary for fuel storage, and as the engine uses liquid fuels, these can be piped by gravity from tank cars to the storage tanks below track level. A 10,000-gallon tank car will supply enough fuel to drive a 100-horsepower engine at full load continuously, day and night, for more than two months. Noncondensing steam engines of the same size consume 4 to 6 pounds of coal or  $2\frac{1}{2}$  to 3 pounds of oil per effective horsepower-hour, or the equivalent of 340 to 500 tons of coal, or 200 to 250 tons of oil as compared with the 40 tons of oil used by the Diesel engine. The saving in storage space is evident.

The labor of unloading and storing coal, firing boilers, and removing the ashes, which may readily amount to one-quarter of the coal fired, is entirely eliminated in Diesel-engine plants. The absence of coal, ash, and smoke makes for cleanliness. The Diesel engine is noiseless and its exhaust is colorless and odorless, so that its operation meets all hygienic requirements.

**FULL POWER QUICKLY AVAILABLE.**

The engine is in readiness for instant operation, and can take on full load from no load as quickly as the operator can make the different shifts, a matter of only two or three minutes. With the stopping of the engine no further fuel is consumed. In this respect the Diesel engine differs from heat engines in which the working fluid is generated outside of the engine—in boilers for steam engines and in gas producers for suction-gas engines. The Diesel engine has no stand-by losses. Boiler or gas-producer losses in firing up, losses up the stack, losses in incompletely burned and clinkered coal, radiation and condensation losses, losses due to inefficient firing of boilers or their insufficient upkeep, or, in suction gas plants, losses due to inefficient producer operation, to afterburning caused by incorrect setting of the valves or timing of the ignition—these add considerably to the increased

heat consumption of steam and gas engines at fractional loads. In this respect the Diesel engine differs from other prime movers in that its over-all fuel economy is not dependent on the efficient operation of auxiliaries, or even on the skill of the operators, as its operation is largely an automatic thermodynamic process in the engine itself.

#### RELIABILITY.

With a correctly designed and efficiently built engine this economy will be maintained undiminished during the life of the engine, provided the engine receives proper care.

The claim of greater reliability for the steam engine is frequently made, because an engine supplied with steam from a boiler will continue to run even if piston and glands leak, or the valves are improperly set for operating economy, and no matter how wastefully the boiler may be operated; whereas the Diesel engine has to generate its own driving fluid in the engine cylinder, and if any part of the engine fails the engine has to stop. However, the supposedly greater reliability of the steam engine is obtained at the expense of a greatly increased fuel consumption.

The Diesel engine has been developed to such a degree of perfection during the past decade that carefully built engines can be considered as being fully as reliable as steam equipment.

No attempt will here be made to outline comparative costs of generating power in the large central stations producing thousands of kilowatts. These unusually large stations, however, are relatively rare, the average size of all central stations in the United States not greatly exceeding 500 kilowatts; and the power demands of most industrial plants are met by small and medium sized plants.

As is outlined subsequently, many industrial plants can generate their own power at less cost than they can purchase current from large central stations.

The simplicity and compactness of Diesel-engine plants is well illustrated by figures 43 and 44 (pp. 100-101), showing complete vertical and horizontal engines.

#### COMPARATIVE EFFICIENCIES OF DIFFERENT PRIME MOVERS.

Table 3 following shows the heat consumption and the thermal efficiencies of different types of prime movers at continuous full load. The high figures of fuel consumption refer to small plants, the low figures to larger plants; they represent good economy in well-operated plants equipped with modern high-grade machinery.

TABLE 3.—*Heat consumption and thermal efficiencies of different types of prime movers at continuous full load.*

| Type of prime mover.                                                                 | Heat consumption per brake horsepower-hour. | Over-all thermal efficiency. | Superiority of Diesel engine. <sup>a</sup> |
|--------------------------------------------------------------------------------------|---------------------------------------------|------------------------------|--------------------------------------------|
|                                                                                      | <i>B. t. u.</i>                             |                              |                                            |
| Noncondensing steam engine <sup>b</sup> .....                                        | 40,000-28,000                               | 6.30-9.1                     | 5.6-3.6                                    |
| Condensing steam engine using superheated steam <sup>b</sup> .....                   | 28,000-16,500                               | 9.1-15.4                     | 3.6-2.3                                    |
| Locomobile engine with superheated steam and reheater, condensing <sup>b</sup> ..... | 17,000-15,200                               | 14.9-16.7                    | 2.4-2.1                                    |
| Steam turbine, superheated steam, 200 to 2,000 horsepower <sup>b</sup> .....         | 24,000-15,500                               | 10.6-16.2                    | 3.2-2.2                                    |
| Steam turbine, superheated steam, 2,000 to 10,000 horsepower <sup>b</sup> .....      | 15,500-14,000                               | 16.2-18.1                    | 2.2-1.95                                   |
| Gas engine without producer.....                                                     | 10,400-9,300                                | 24.4-27.5                    | 1.33-1.28                                  |
| Suction gas engine <sup>c</sup> .....                                                | 14,000-11,200                               | 18.1-22.7                    | 1.95-1.55                                  |
| Diesel engine.....                                                                   | 8,000-7,200                                 | 32-35.3                      | .....                                      |

<sup>a</sup> Figures in this column are to be used as factors with which to multiply values in preceding column.<sup>b</sup> Figures include boiler losses.<sup>c</sup> Figures include producer losses.

## COMPARATIVE COST DATA.

## COST OF OPERATION.

Table 4 following presents fuel-cost data covering the operation of the prime movers represented in Table 3:

TABLE 4.—*Comparative operating costs of different types of prime movers.*

| Type of prime mover.                                                                 | Kind of fuel.                                         | Average cost per 1,000,000 B. t. u. | Cost of 1,000,000 B. t. u., effective work. | Heat cost per one effective horsepower-hour. |
|--------------------------------------------------------------------------------------|-------------------------------------------------------|-------------------------------------|---------------------------------------------|----------------------------------------------|
|                                                                                      |                                                       | <i>Cents.</i>                       | <i>Cents.</i>                               | <i>Cents.</i>                                |
| Noncondensing steam engine <sup>a</sup> .....                                        | Coal.....                                             | 12                                  | 191-132                                     | 0.48-0.34                                    |
| Condensing steam engine using superheated steam <sup>a</sup> .....                   | .....do.....                                          | 12                                  | 132-78                                      | .34-.20                                      |
| Locomobile engine with superheated steam and reheater, condensing <sup>a</sup> ..... | .....do.....                                          | 12                                  | 81-72                                       | .21-.18                                      |
| Steam turbine, superheated steam, 200 to 2,000 horsepower <sup>a</sup> .....         | Coal and anthracite.....                              | 12                                  | 113-74                                      | .29-.19                                      |
| Steam turbine, superheated steam, 2,000 to 10,000 horsepower <sup>a</sup> .....      | .....do.....                                          | 11                                  | 104-68                                      | .27-.17                                      |
|                                                                                      | .....do.....                                          | 12                                  | 74-67                                       | .19-.17                                      |
|                                                                                      | .....do.....                                          | 11                                  | 68-61                                       | .17-.155                                     |
| Gas engine without producer.....                                                     | Natural gas, coke-oven gas, or blast-furnace gas..... | 15                                  | 62-55                                       | .16-.14                                      |
|                                                                                      | .....do.....                                          | 7                                   | 27                                          | .7                                           |
| Suction gas engine <sup>b</sup> .....                                                | Anthracite.....                                       | 11                                  | 61-49                                       | .16-.14                                      |
| Diesel engine.....                                                                   | Petroleum.....                                        | 15-18                               | 56-43                                       | .14-.11                                      |

<sup>a</sup> Figures include boiler losses.<sup>b</sup> Figures include producer losses.

## COST OF FUEL.

Cost data covering the various types of fuel used in the prime movers represented in Tables 3 and 4 are presented in Table 5 following:

TABLE 5.—*Cost of various types of fuel.*

| Kind of fuel.             | Price of fuel.                             | Heating value per pound. | Absolute heat cost per 1,000,000 B. t. u. | Average heat cost per 1,000,000 B. t. u. |
|---------------------------|--------------------------------------------|--------------------------|-------------------------------------------|------------------------------------------|
|                           |                                            | <i>B. t. u.</i>          | <i>Cents.</i>                             | <i>Cents.</i>                            |
| Lignite.....              | \$1.00 to \$2.50, ton of 2,000 pounds..... | 5,000-9,000              | 10-14                                     | 12                                       |
| Bituminous coal.....      | 2.00 to 4.00, ton of 2,000 pounds.....     | 11,000-14,200            | 9-14                                      | 12                                       |
| Anthracite.....           | 2.50 to 4.00, ton of 2,000 pounds.....     | 14,500                   | 8.6-13.8                                  | 11                                       |
| Fuel oil (petroleum)..... | .75 to 2.25, barrel.....                   | 18,000-19,000            | 12.5-37.5                                 | 15-18                                    |
| Natural gas.....          | .10 to .75, 1,000 cubic feet.....          | 900-1,000                | 11-75                                     | 15                                       |
| Blast-furnace gas.....    | .05 to .01, 1,000 cubic feet.....          | 90                       | 5.5-11                                    | 7                                        |
| Coke-oven gas.....        | .02 to .05, 1,000 cubic feet.....          | 450                      | 4.4-11                                    | 7                                        |



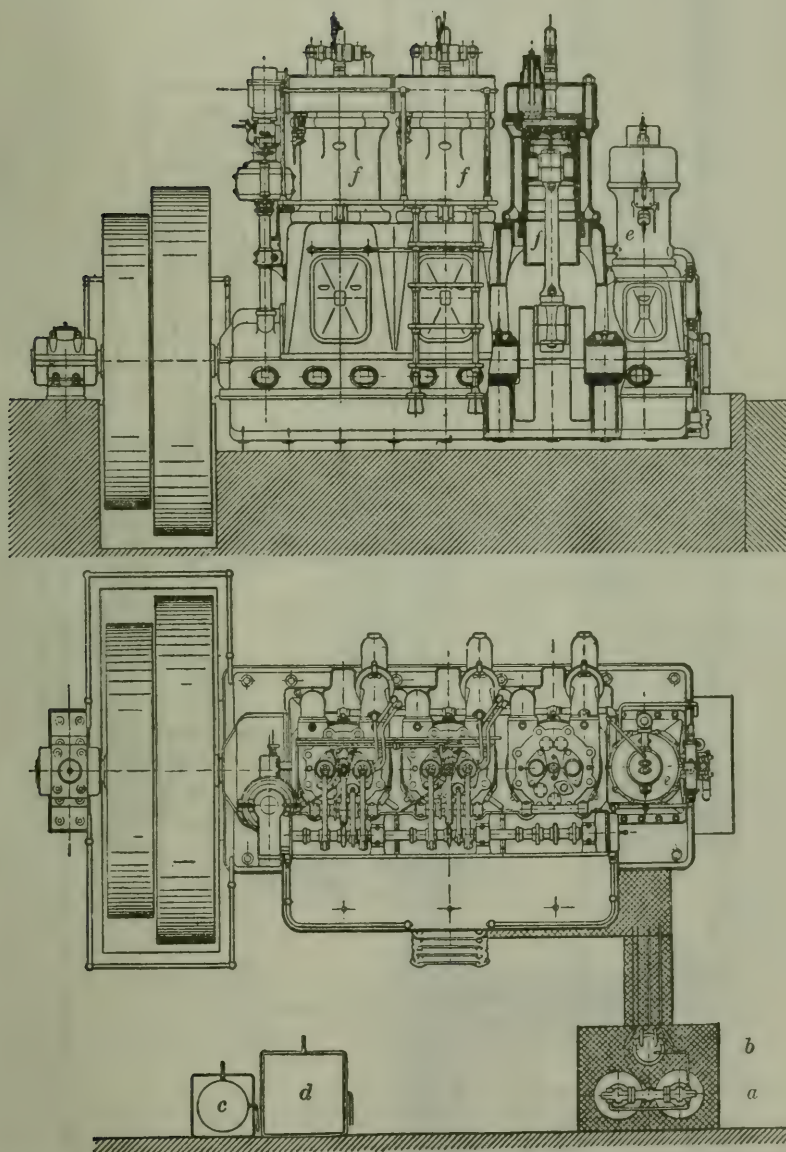


FIGURE 42.—Plan and elevation of vertical three-cylinder Diesel engine with all auxiliaries: *a*, air receiver; *b*, injection air bottle; *c*, ignition-oil tank; *d*, fuel-oil tank; *e*, air compressor; *f*, working cylinders.

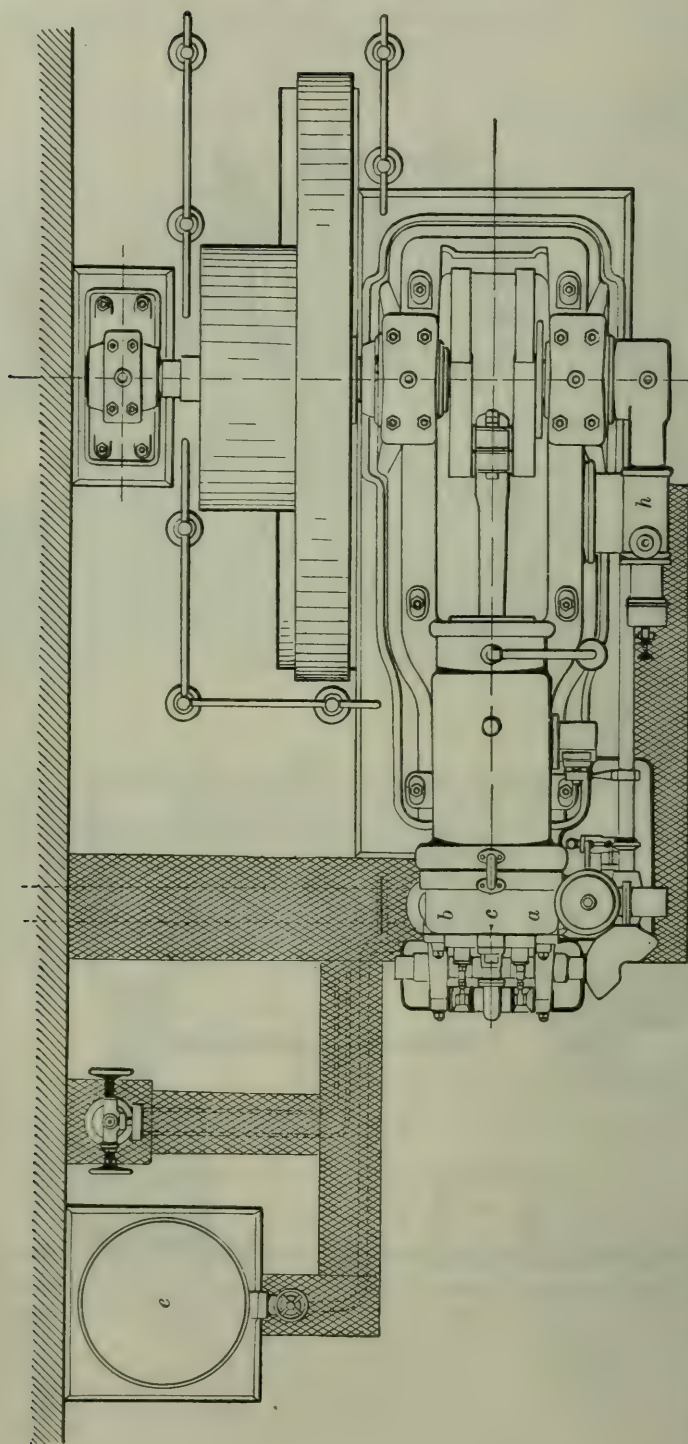


FIGURE 43.—Plan of horizontal Diesel engine with all auxiliaries. *a*, air valve; *b*, exhaust valve; *c*, fuel valve; *d*, starting valve; *e*, receiver for starting air; *f*, receiver for injection air; *g*, fuel tank; *h*, air compression.

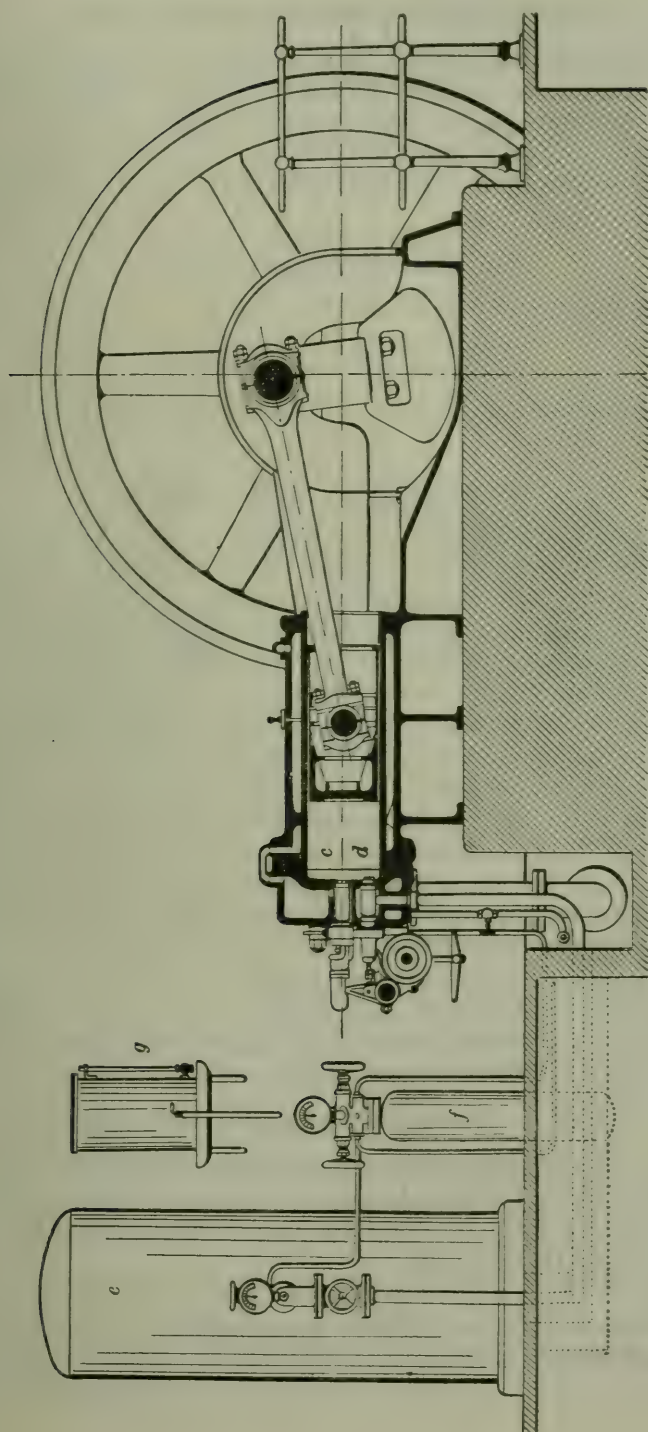


FIGURE 44.—Elevation of engine shown in figure 43.



**FORMULAS FOR COMPUTING FUEL COST.**

Fuel is usually the principal item of expense in generating power. Because of the great variety of fuels, and their differing greatly in heating value and other properties, all fuel prices should be reduced to a common basis of absolute cost per 1,000,000 B. t. u. The cost of 1,000,000 B. t. u. actually converted into mechanical work is then :

$$\frac{\text{Cost of 1,000,000 B. t. u.}}{\text{Over-all thermal efficiency of plant.}}$$

The fuel cost per effective horsepower-hour is

$$\frac{\text{Cost of 1,000,000 B. t. u.} \times 2,545.}{\text{Over-all thermal efficiency} \times 1,000,000.}$$

In this expression, 2,545 B. t. u. is the heat equivalent of 1 English horsepower per hour.

**SIGNIFICANCE OF COST DATA.**

Tables 4 and 5 (p. 98) give the fuel prices and heat values of different kinds of fuels, the absolute heat cost of 1,000,000 B. t. u. and the cost of 1,000,000 B. t. u. converted into mechanical work. The heat cost of 1 horsepower-hour has also been computed for the different kinds of engines, based on average fuel prices in widely different sections of the United States. The figures mentioned indicate that although the absolute fuel price for fuel oil may be several times higher than that of coal in the same market, power may be generated more cheaply with the Diesel than with the steam engine. However, the figures given are representative of continuous full-load conditions, at which power plants seldom operate. At fractional loads, the economy of the steam engine decreases; there is also additional fuel consumed to make up for boiler and producer losses when firing up or banking fires, and there are stand-by and radiation losses which increase materially with fractional loads. In the Diesel engine the fuel consumption increases only in proportion as the fractional load increases the internal work of the engine, with a decrease in its effective output. When the engine stops all fuel losses stop.

**EFFECT OF DIFFERENT FACTORS ON ECONOMY.**

How the size of units affects their economy is shown by figure 45. The Diesel engine has the highest fuel economy, which, moreover, is least affected by the size of the unit. The economy is nearly constant for all Diesel engines.

In figure 46 the average increase in fuel consumption with fractional loads is shown for the different prime movers, based on actual engine performance in well-operated plants. To obtain the economy

of steam plants at fractional loads, the full-load economy for a given size of plant can be taken from figure 45 and then the percentage increase computed for that size from figure 46.

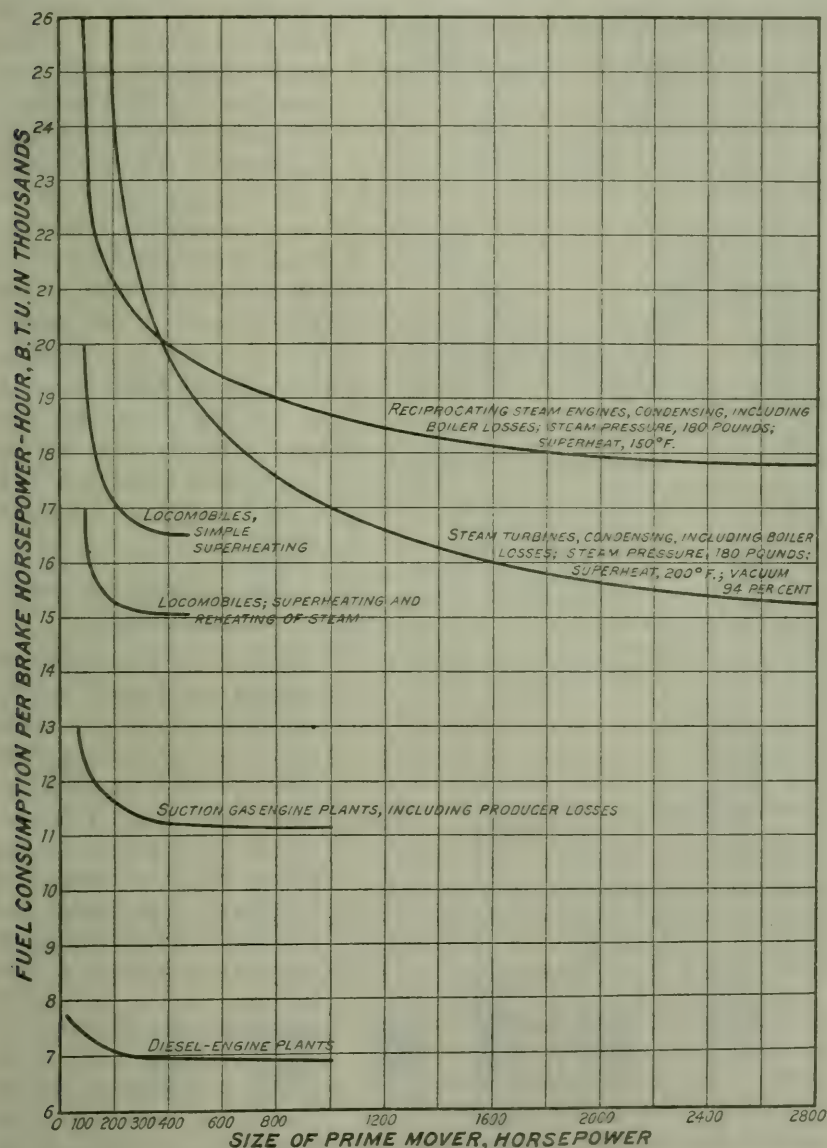


FIGURE 45.—Influence of size on efficiency of different prime movers at continuous full load.

With inexperienced labor and insufficient care of machinery, the fuel consumption of the steam and the suction gas power plants may be still higher than is shown in the figures.

From the data that has been presented it is evident that small steam plants can not compete with Diesel engines in the cost of generating power. On the basis of results of the actual operation of steam and of

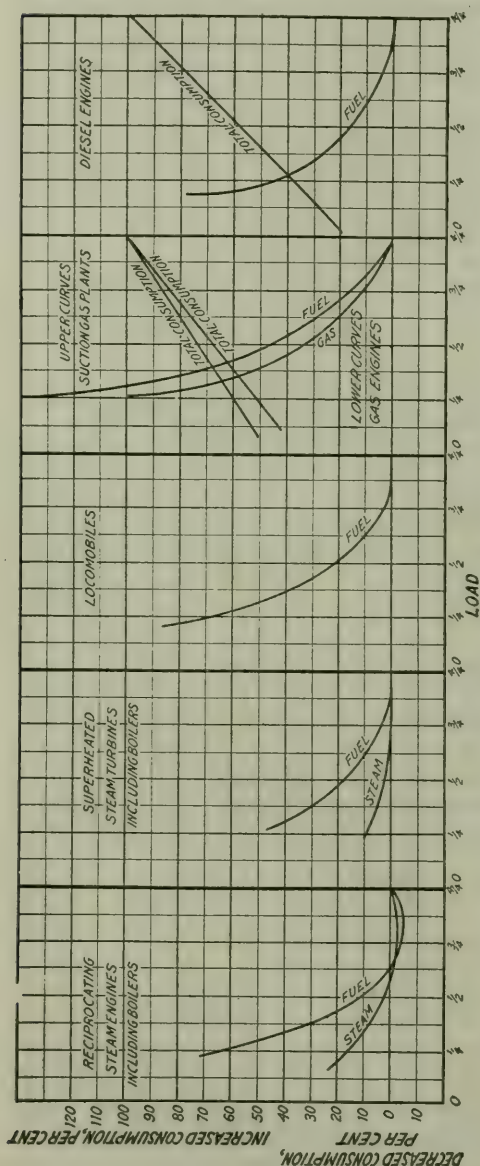


FIGURE 46.—Average increases in fuel consumption by different prime movers at fractional loads.

modern Diesel-engine plants, allowance being made for the special factors favoring the Diesel engine, such as small space requirements, constant readiness for immediate operation, cleanliness, and absence of danger from possible boiler explosions, steam plants up to 1,000 horsepower, using coal costing 10 cents per 1,000,000 B. t. u. can not effectively compete with Diesel-engine plants even with fuel costing 30 cents per 1,000,000 B. t. u.

#### ADVANTAGES OF LOCOMOBILES.

Among the types of steam engines considered, the locomobiles occupy a place apart, on account of their phenomenal economy for their small size. The locomobile is a self-contained power plant, consisting of a boiler, a superheater, and a compound condensing engine, a condenser, a soot blower, and a feed-water heater. It is usually equipped also with a reheater in which the steam is reheated before it is admitted to the low-pressure cylinder. These units are

widely used in Germany and other parts of Europe, and in recent years an American concern has begun to manufacture them.

In actual everyday operation, however, these units do not meet the performance guaranties established by test. A recent investiga-



tion of the performance of 12 locomobile plants and 15 Diesel-engine installations in different parts of Germany disclosed the fact that during one year's operation with the changing load conditions common

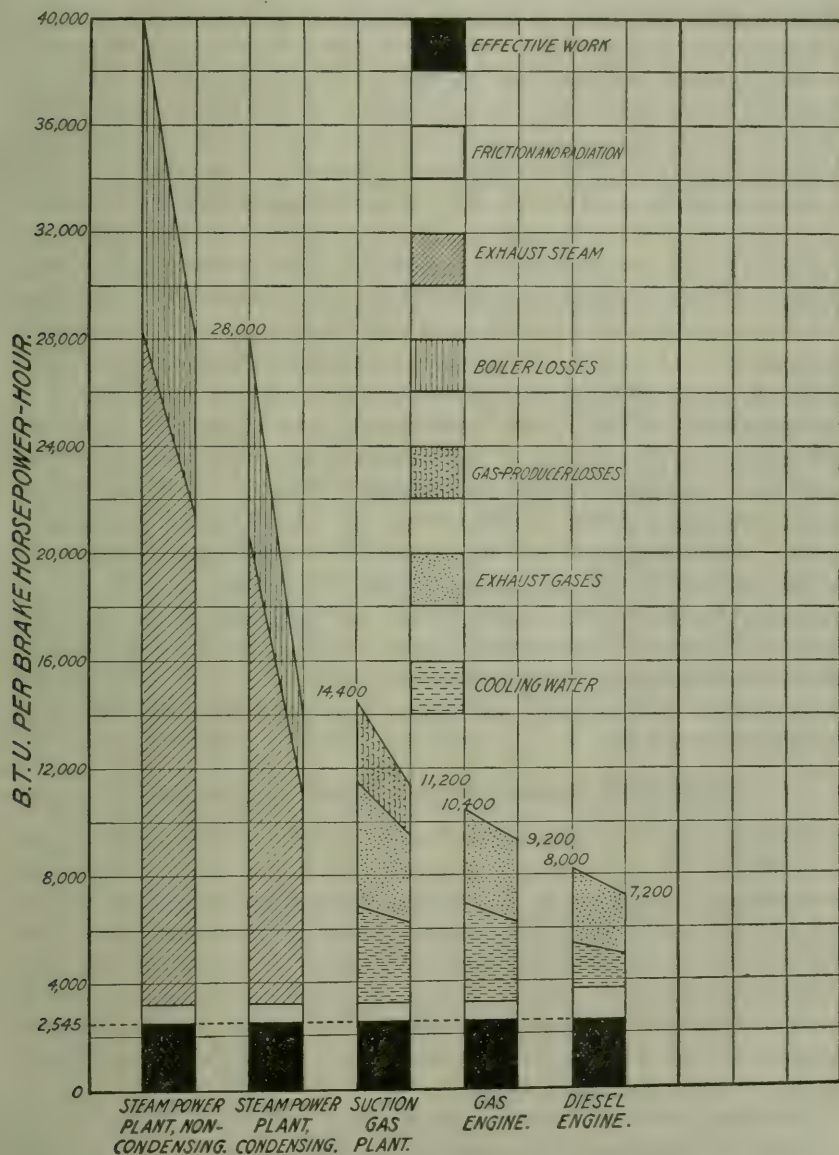


FIGURE 47.—Heat balances of different kinds of engines.

to most central stations the actual fuel consumption of the locomobile plants averaged 102 per cent higher than the guaranteed performance, whereas the consumption of the Diesel-engine plants was only 14 per cent higher than the performance guaranteed.

**LIMITATIONS OF SUCTION GAS PLANTS.**

The locomobile units, on account of their economy, may be used advantageously in sections of the country where fuel oil is not obtainable or where the price is high as compared with the price of coal. Similarly, a suction gas plant may be in order where cheap anthracite or coke is to be had and fuel oils are so high in price as to offset the manifold advantages and the superiority of the Diesel engine over suction gas plants. With the use of bituminous coal, the operating difficulties of suction gas engine plants increase, and not all bituminous coals can be used successfully in suction gas producers. Success depends greatly on the experience of the operator in making a uniform gas of the desired composition. Further, the valve setting and the timing of ignition must be adjusted to the composition of the gas. On account of a lack of experienced labor to operate such plants, disinclination on the part of users to pay advanced wages for such experience, or, possibly on account of deficient design of the producer or the gas engine or both, many such plants have failed, and they have not been as widely adapted as their relatively high fuel economy would lead one to expect. With the present development of the Diesel engine, a further decline in the use of suction gas engine plants may follow. Suction gas engine plants cost fully as much as Diesel-engine plants per horsepower of capacity; they take up more room for equipment and for fuel storage, and depend on the producer for their gas supply. Their fuel consumption under average operating conditions, even in well-conducted plants, is seldom better than  $1\frac{1}{4}$  to  $1\frac{1}{2}$  pounds per horsepower, compared with about one-half pound of fuel oil for Diesel-engine plants.

**CONDITIONS MAKING OTHER PLANTS DESIRABLE.**

When a supply of natural gas, coke-oven gas, or blast-furnace gas is available, the gas engine is generally the logical selection. If the heat price of these gases as well as the load factor of the station is very low and the load is highly fluctuating, large steam-turbine plants are the correct economical selection at present, as they cost least per kilowatt of capacity. The reduction in capital cost overbalances any fuel saving possible with gas engines, this saving being relatively small because the absolute fuel consumption for the limited kilowatt-hour output of such stations is small.

The selection of the steam engine in small units when power is to be transmitted electrically to motors is justified only when the exhaust steam can be used either for heating buildings or in industrial processes.

The steam turbine has the advantage of lower installation cost, smaller space and foundation requirements, and oil-free steam. The steam consumption of small turbine units is, however, materially

higher than that of steam engines; their use is therefore warranted only when the amount of heat to be supplied by the exhaust steam is very large.

Figure 47 shows the comparative value of the steam engine for heating purposes. This figure shows the distribution of the amount of heat wasted per horsepower-hour after the amount required for conversion into mechanical work (2,546 B. t. u.) has been deducted. In condensing steam engines, 12,000 B. t. u. per horsepower-hour and in noncondensing steam engines 20,000 to 24,000 B. t. u. per horsepower-hour are available in the exhaust steam for heating purposes, whereas in gas and Diesel engines only about 2,000 and 1,200 B. t. u. per horsepower-hour can be abstracted from the waste gases. If the waste gases are utilized for heating, the steam engine generates power cheaply, the fuel chargeable to power production being reduced to about 8,000 to 16,000 B. t. u. per horsepower-hour.

The cooling water and the waste gases from medium and large sized Diesel engines can be used for heating when only a limited supply of hot water is needed. The safety of the engine precludes heating of the cooling water to a temperature higher than 160° F. In large engines, to avoid excessive heating of cylinders and cylinder heads, the cooling water should not be discharged at a temperature higher than 125° F. If the exhaust gases are used to heat this water (of which about 5 gallons per horsepower-hour is needed), still further, in specially constructed heaters or boilers, the efficiency of the engine drops off rapidly with an increase in the temperature to which the water is heated, as does the volume of water so heated. Five gallons per horsepower-hour may be thus heated from 125° to 170° F., and only 2½ gallons if heated to 212° F. The volume of water that can be heated is roughly reduced in proportion to a reduction in the indicated power of the engine. The Diesel engine is therefore used principally for generating power.

#### FACTORS AFFECTING COST OF GENERATING POWER.

As regards the generation of power with heat engines the following factors influence the cost:

1. The fuel consumption of the generating units.
2. The B. t. u. price.
3. The size of generating units and of the station.
4. The station-load factor or the ratio of yearly output in kilowatt-hours to the maximum possible yearly output in kilowatt-hours.
5. Capital or fixed charges, comprising interest on the capital invested in the power plant and amortization.
6. Operating expenses comprising cost of labor, lubrication, and maintenance.



The influence of these factors on the cost of generating power with small Diesel engines is shown in figures 48 and 49. One hundred horsepower and 200-horsepower engines which drive industrial machinery direct from belt have been selected. The engines are loaded at an average of about 80 per cent of their rated capacity, a condition commonly found in industrial plants. Labor is figured at \$3 per man per 8-hour day. The installation cost, including building space for housing the engine, is taken at \$100 per horsepower of installed capacity for the

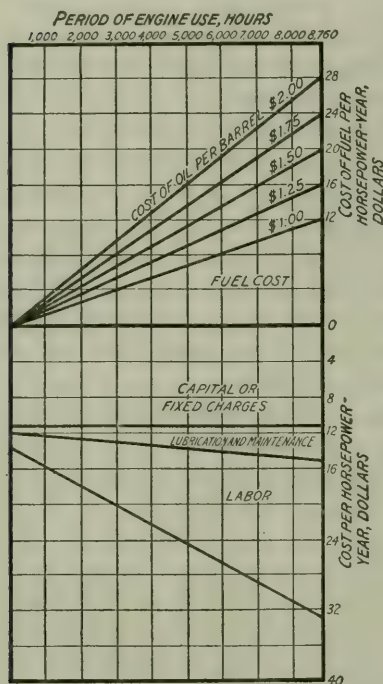


FIGURE 48.—Comparative costs of different items in the operation of a 100-horsepower Diesel engine at a mean 80 per cent load.

100-horsepower engine and at \$90 for the 200-horsepower engine. All costs are referred to a unit of 1 horsepower operating one year of 8,760 hours. The total costs of operating the engine for shorter periods are also shown. The figures show that capital charges are constant, regardless of load; that the fuel cost is proportionate to the operating time; and that the labor and the maintenance charges are nearly proportionate to the operating time.

The cost graph of the 200-horsepower engine (fig. 49) shows the influence of size on the reduction of capital and of labor charges. There is no difference in the fuel consumption of the two engines, as

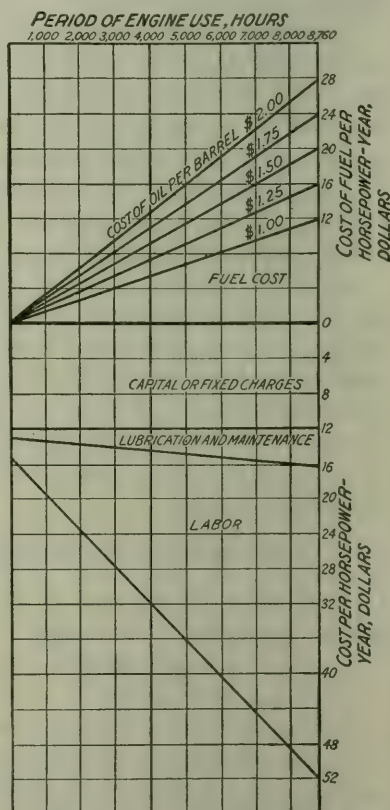


FIGURE 49.—Comparative costs of different items in the operation of a 200-horsepower Diesel engine operating at a mean 80 per cent load.

the fuel economy of both is the same. Figure 50 shows for the 100-horsepower engine the operating costs for different engine-use periods reduced to costs per horsepower-hour corresponding to these use periods or load factors. Figure 51 shows similar data for the 200-horsepower engine. The influence of a reduced number of operating

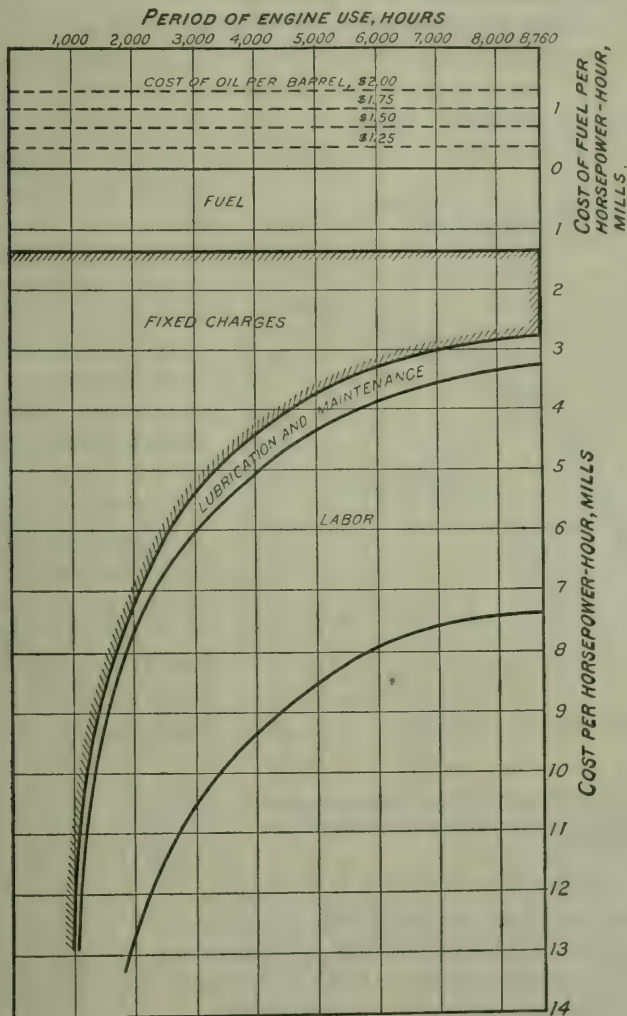


FIGURE 50.—Operating costs for different engine-use periods reduced to corresponding horsepower-hour costs, 100-horsepower Diesel engine.

hours, which is synonymous with a reduced output of horsepower-hours, is at once apparent. The fixed charges increase the unit power cost rapidly at low-use factors. For the fuel cost a basic price of \$1 per barrel of 320 pounds has been used. To what extent higher fuel prices increase the power cost is indicated by broken lines

above the base line. The cost increment proportionate to the higher fuel price would have to be added to the horsepower-hour cost shown in the graph below the base line (in which the fuel cost at \$1 per barrel is already included) for any use factor. As this cost increment is the same for both engines, figure 49 includes only the fuel cost based on oil at \$1 per barrel.

Figure 53 shows the influence of the size of Diesel-engine equipments for generating electric power on the cost of producing the power. The capital charges are based on the costs per unit of installation as given in figure 52. To obtain the best operating economy and flexibility, each station is considered as being com-

posed of at least three units, each with a capacity of one-third of the aggregate station capacity. With an increase or a decrease in load during different periods of the day one or more units can be thrown into or out of action.

Figure 54 shows the cost of a kilowatt-hour for different station-load factors and for stations of different sizes, the data being based on the kilowatt-year costs in figure 53. The fixed charges are represented by full lines marked KW. F. C., and the costs for labor, lubrication, and maintenance are represented

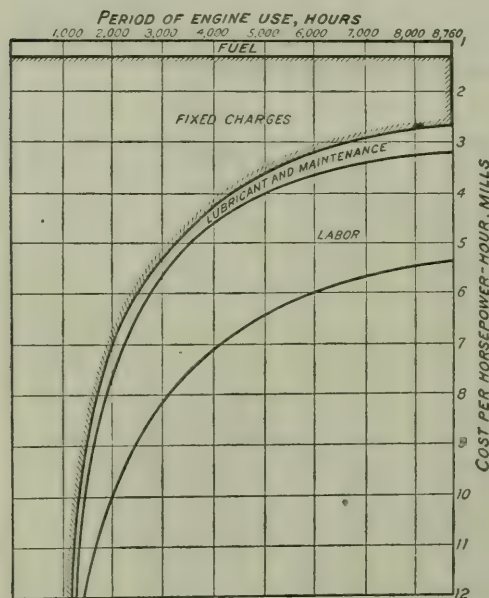


FIGURE 51.—Operating costs for different engine-use periods reduced to corresponding horsepower-hour costs, 200-horsepower Diesel engine.

represented by broken lines marked KW. L. M. The fuel costs have not been added, as these are practically constant at 2 to 2½ mills per kilowatt-hour for a load factor of 100 to 25 per cent, with fuel oil costing \$1 per barrel of 320 pounds. For oil of a higher price the proportionate higher fuel cost can easily be calculated.

Combined charges for capital, labor, lubrication, and maintenance can be read direct for any load factor from the broken-line curves. The line representing kilowatt-hour costs for a 6,000-kilowatt station has been cross-hatched to make it distinct. As the size of a station materially influences the cost of power production, different sized stations are indicated by separate curves to make clear the influence of station size and station factor on the power production cost.



Figure 55 presents a comparative analysis of the factors controlling the cost of generating power in medium-sized (9,000-kilowatt) central stations using either Diesel engines or steam turbines. Figure 56 shows the comparative efficiencies of the two types of prime movers. For the steam turbine highly favorable operating conditions have been taken, namely a steam pressure of 180 pounds per square inch; a steam temperature of 600° F.; a temperature of 60° F. at the inlet for the circulating water for condensing, and a vacuum of 96 per cent.

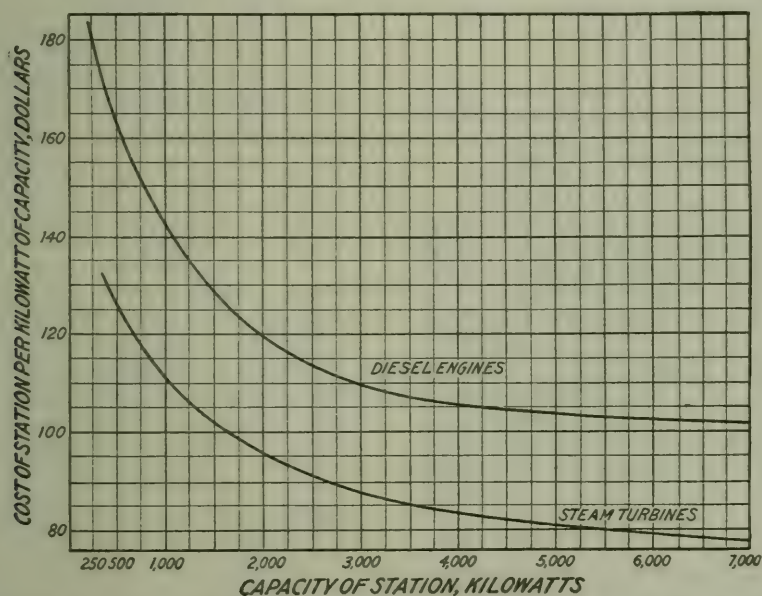


FIGURE 52.—Comparative costs per unit of Diesel-engine equipment and of steam-turbine equipment for generating electricity.

Of the three variables, steam pressure, steam temperature and cooling-water temperature, the last two affect the economy most, as is shown in figure 57, from which the increased or decreased fuel consumption can be figured for other operating conditions than those assumed. The percentage of fuel increase (above the base line marked "cooling-water temperature") and the percentage of fuel decrease (below the base line marked "cooling-water temperature") to be added to or to be deducted from the performance consumption used as a starting point can be readily ascertained for different sets of operating conditions.

Figure 55, showing the total yearly capital, lubricating, maintenance, and fuel charges for different station factors, indicates that the fuel price and the station factor (relation between average load and rated capacity) determine which type of plant is the more economical for a given set of conditions.

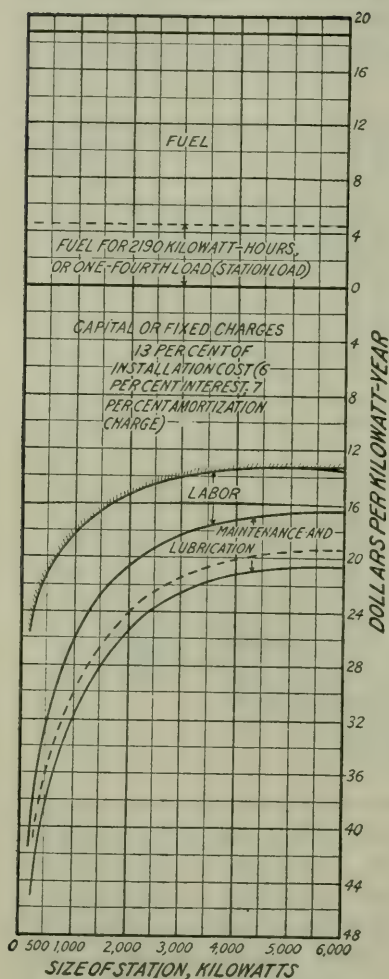


FIGURE 53.—Curves showing cost of kilowatt-year (8,760 kilowatt-hours) at full load, as influenced by station capacity.

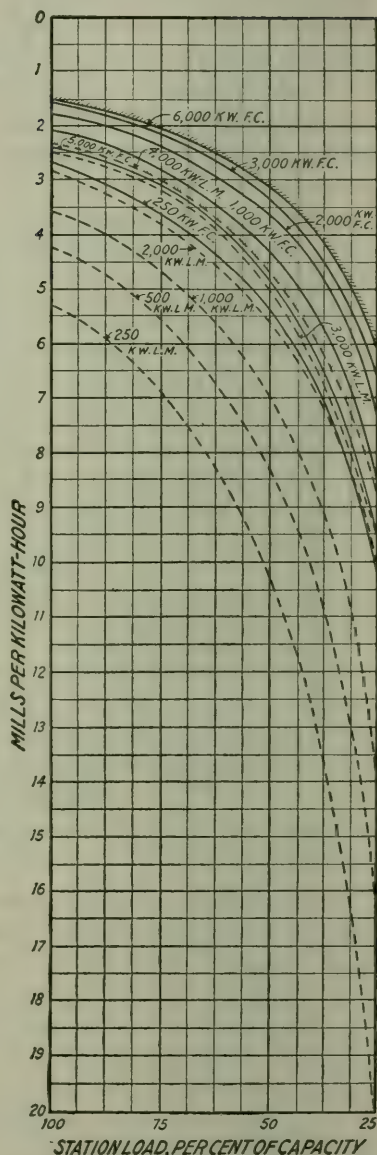


FIGURE 54.—Curves showing cost of kilowatt-hour in mills at different station load factors. Fuel cost per kilowatt-hour is 2 to 2½ mills, which must be added to costs shown.

## SUMMARY OF COMPARATIVE ADVANTAGES.

Following is a statement of the comparative advantages and disadvantages of Diesel engines and of steam turbines for generating power:

*Summary of comparative advantages and disadvantages of Diesel engines and of steam turbines for generating power.*

## ADVANTAGES OF STEAM-TURBINE PLANTS.

1. Unlimited size of generating units.
2. High rotative speed, reducing size of generating units, and cheapening turbine and alternator construction.
3. Low installation cost.
4. High overloading capacity.
5. Ability to use under the boiler all classes of fuels (solid, liquid, or gaseous), including the cheaper varieties of inferior-grade coals.

## DISADVANTAGES OF STEAM-TURBINE PLANTS.

1. The fuel economy is greatly dependent on the skill of the operating force and the degree of thoroughness with which the installation is maintained at operating efficiency.
2. Dangers and drawbacks of boiler operation.
3. Boiler losses generally and fuel losses in firing up and in banking boilers and in ash and clinkers.
4. Exacting requirements relating to circulating water for condensing the steam, one of the deciding factors in the commercial success of a steam plant.

## ADVANTAGES OF DIESEL-ENGINE PLANTS.

1. Very high thermal efficiency and lower absolute heat rate, independent of the size of unit.
2. Absence of all boilers, smoke, and soot.
3. Constant readiness for operation.
4. No fuel losses in firing up and in unconsumed coal in ash as in steam plants.
5. Small volume of cooling water required.
6. Use of cheap liquid fuels, such as petroleum residues, coal-tar oils, and certain coal tars.

## DISADVANTAGES OF DIESEL-ENGINE PLANTS.

1. Limited size of units (at present, 4,000 horsepower).
2. Massive foundations necessary.
3. High consumption of lubricating oil.
4. High installation cost.

The fuel costs are figured for different B. t. u. prices, ranging from 10 to 20 cents per 1,000,000 B. t. u. Those represented in the right graph of figure 55 are based on the use of coal under the boilers. As higher boiler efficiencies are obtained with fuel oil than with coal, and there is also less labor involved in caring for boilers, allowance for a decrease in cost for a steam-turbine plant using fuel oil should be made comparing such a plant with a Diesel-engine plant on the basis of the data represented in the figure. Thus, the next lower price than that shown for a given fuel cost can be taken. For instance, if fuel oil costs 16 cents per 1,000,000 B. t. u., take the cost given for oil at



14 cents as representing the fuel cost for the turbine plant and compare it with the 14-cent price for the Diesel-engine plant.

### CONSIDERATIONS GOVERNING CHOICE OF PRIME MOVERS.

The data represented in figure 53 indicate that the selection of either type of prime movers should be governed by the following economic considerations:

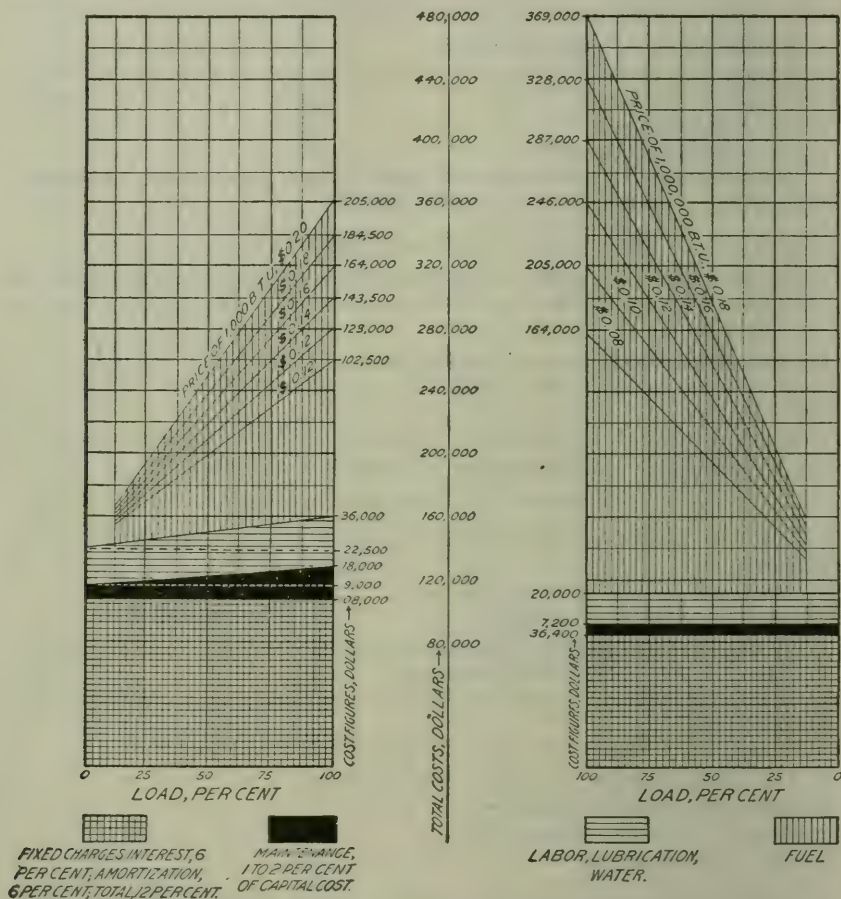


FIGURE 55.—Comparative costs of operation of Diesel-engine and of steam-turbine installation for generating power. Left graph represents four 3,500-horsepower Diesel engines in a plant costing complete, including buildings, \$900,000; right graph represents three 3,000-kilowatt steam turbines in a plant costing complete, including buildings, \$720,000.

1. Fuel is of chief influence on the total cost of power when both the price per 1,000,000 B. t. u. and the load factor are high. These conditions favor the use of the Diesel engine.

2. Interest and amortization charges are of chief influence on the total power cost with low B. t. u. price and low load factor, particularly as applied to stand-by plants, which are operated only occasion-

ally or to take care of recurring peak loads. The installation cost of such plants must be kept as low as possible so as to avoid heavy capital charges distributable over a relatively small output of kilowatt-hours for the station. Such conditions favor the steam turbine.

3. An exception to consideration 2 is when the constant readiness of the Diesel engine makes it preferable, installation cost being of secondary importance. Here the cost of keeping boilers under steam continuously would have to be balanced against the difference of interest charges on steam-turbine plants as compared with those on Diesel-engine plants.

4. The prime movers most logical for power plants situated at the source of the fuel, either in the oil field or at the coal mine, and therefore enjoying the advantages of a cheap fuel supply, are those costing least to install—that is, steam turbines. The fuel expenditures will be comparatively less than interest and redemption charges, provided that a satisfactory supply of cooling water is available for condensing.

5. In many instances combination plants using Diesel engines for supplying the continuous and nearly constant main load, and steam turbines for furnishing periodically occurring peaks by the use of high-duty boilers with large water and steam spaces, capable of being forced when necessary, will prove most profitable. The periodical peaks may be taken care of by a turbine floating on the line and operating in parallel with the Diesel engines that supply the main load and operate constantly at or near full load.

6. Up to capacities of 1,000 horsepower, steam turbines can compete with Diesel engines only for special uses, such as supplying exhaust steam for heating. For larger plants, of 1,000 to 10,000 kilowatt capacity, careful analysis must be made of the relative advantages of Diesel engines and of turbines, a knowledge of the load factor, the fuel prices, and the water supply being necessary.

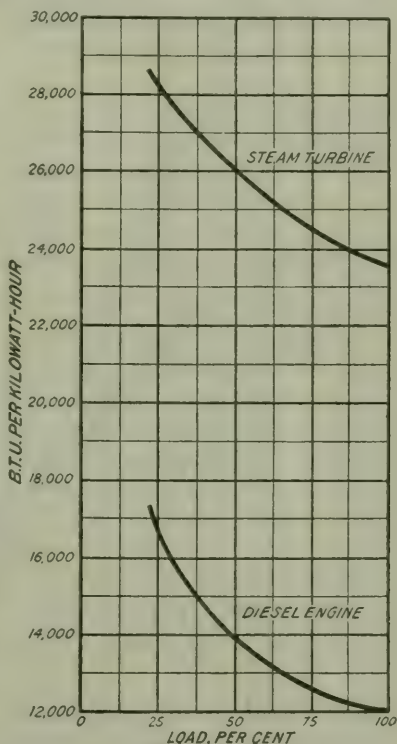


FIGURE 56.—Comparative operating efficiencies of Diesel engines and of steam turbines. Steam-turbine equipment to generate 3,000 kilowatts; figures include boiler losses and operation of auxiliaries; steam pressure, 180 pounds per square inch; steam temperature, 600° F.; temperature, at inlet, of circulating-water for condensing, 60° F.; vacuum, 96 per cent. Single-acting, two-stroke, vertical, Diesel-engine equipment to generate 3,500 horsepower; 125 revolutions per minute; 3,000-kilowatt generator.

For power plants with a capacity larger than 10,000 kilowatts, comprising units of 6,000 kilowatts or more, steam turbines are preferable, unless a combination of high load factor, high fuel cost, and unfavorable water supply favors the Diesel engine. This combination is not frequently encountered.

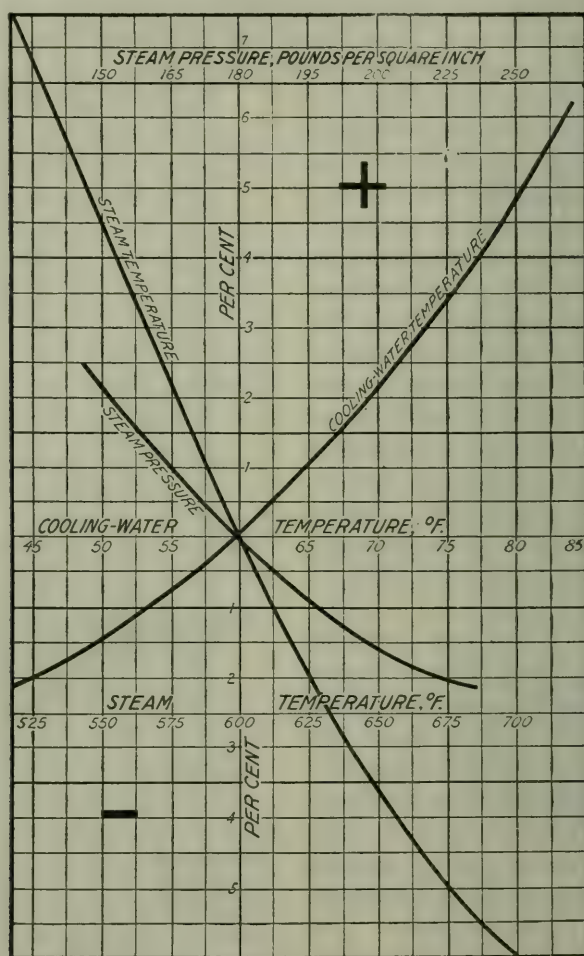


FIGURE 57.—Influence of operating conditions on performance.

### SPECIAL ADAPTATIONS OF DIESEL ENGINES.

Mention has been made of combination plants in which steam turbines and Diesel engines are used. Great economies are possible with such plants. As steam turbines are flexible, can be greatly overloaded, and are relatively cheap per kilowatt of capacity, they can be used to furnish peak loads of short duration. Such loads occur in almost every central station in the early night hours, when the maximum demand on the station may be several times the



average demand. Although the steam unit consumes considerably more fuel, the absolute total fuel consumption of the steam equipment for the kilowatt-hour output at the peak is comparatively small, as the Diesel engine would be running at full-load capacity during the maximum current output of the station. During the hours in which the steam turbine is not in service, the exhaust gases and the discharged cooling water from the Diesel engine would be used for keeping the boiler heated and under steam. The use of the exhaust gases for this purpose is, of course, applicable only to the moderately large station.

Many small steam plants now in operation can make appreciable fuel economies by continuing the use of the present steam equipment for supplying the peak load and by installing sufficient Diesel-engine equipment to provide the average load. The most economical size and number of Diesel-engine units in relation to existing steam equipment can be determined only on the basis of the average daily load for a period of a year, and each power problem has to be considered separately. A practical example of the savings possible is of interest. A fluctuating yearly load of 1,250,000 kilowatt-hours was supplied by a 500-kilowatt steam turbine, operating on the condensing system, with an average fuel consumption of nearly  $6\frac{1}{2}$  pounds of coal per kilowatt-hour. By the installation of a 200-kilowatt Diesel engine unit, at a cost of \$30,000, a yearly fuel saving of \$11,000 could have been realized. In this instance the coal was high priced, costing nearly 20 cents per 1,000,000 B. t. u. Such a degree of saving is possible in steam plants all over the United States, the fuel saving paying for the Diesel-engine equipment in three to four years without necessitating any increase in labor for operating such mixed plants.

Analysis of the different factors controlling the cost of generating power with Diesel engines shows clearly that, in common with other types of power plants, this cost diminishes with an increase in the size of the plant. The reduction in power-generating costs is due, first, to decreased interest and redemption charges for larger plants, these charges being in proportion to the capital outlay for the plant, and, second, to a lessened labor charge. A possible conclusion is that large central stations must produce the power cheaper than smaller independent and isolated engine units. However, the influence of the station factor on the power cost was also made evident. Furthermore, the figures given for cost of generating power were for current delivered at the station switchboard. They did not include the cost of distributing the current, which is often a more expensive operation, when the interest and redemption charges on transmission and distributing systems and the cost of power losses are taken into account.

The cost of transmission and distributing systems and the expense of their maintenance depend on such a variety of conditions that no generally applicable cost data can be given, but each system must be designed for the particular conditions to be met.

There are, however, large fluctuations in the capital cost (including interest charges and amortization) of different systems per kilowatt of capacity, and, what is more important, per kilowatt-hour of delivered energy. This cost is greater for a thinly settled distributing area and, if there are few power consumers in the area, the transmission line must be comparatively longer for each kilowatt of used output, and the power consumption will be more irregular. The power losses of such systems are also considerable. These factors all combine to make hydraulic power by no means as universally cheap as many people believe. Fixed charges resulting from the high capital cost of some of the large hydraulic-power and electric-transmission systems, and the unavoidable power losses of such systems, add so much to the total cost of delivered energy that Diesel-engine plants can often generate power more cheaply than can hydroelectric systems. The advantage of the Diesel engine is especially apparent if it can be applied direct without the double transformation of energy—mechanical into electrical and this back to mechanical. The Diesel engine is also preferable for central stations supplying current within a limited area, as the necessity of high-tension long-distance transmission is obviated. It is well recognized that large power companies frequently are obliged to sell current to large consumers at cost, or even below cost, in order to create a demand for more power. The increase in power output insures a more favorable load factor, but the numerous small consumers have to pay high prices to make up for the losses in profit on the power supplied to the large consumers.

The great economy of the Diesel engine makes even moderate-sized central stations for small cities and towns and isolated power plants feasible and renders possible the generation of such cheap power that the large central stations with overland-transmission systems can compete with them only in densely settled areas.

Most industrial plants that operate day and night, and therefore have an exceptionally high load factor, can generate their power with Diesel engines at such a cost, including all operating and fixed charges (12 to 15 per cent), that they can fix a profitable price that no central station can equal even in powers as small as 100 horsepower. Even with a 50 per cent load factor, such an engine will, as a rule, generate power cheaper than it can be purchased.

Of late years, especially in Europe, much has been done to develop the oil engine in small powers, to take the place of the electric motor. There is a great demand for such small-powered engines, particularly in agriculture. There is also a large field for small engines of

this type for tractors and motor trucks. The great weight of the Diesel engine and the first high cost per horsepower of capacity was against their adoption in the earlier stages of their development. Likewise, the necessity for thorough vaporization of the fuel was not so well understood, and the use of a compressor for the fuel injection made the cost of the small units excessive, not to mention the great difficulties of building so small an air compressor that would be reliable with the high pressures used. The clearance space and the valves would be so small that they would be fouled by a minute quantity of dirt and oily deposit.

For small powers a highly successful type of engine has been developed, in which the measured fuel charge is deposited during the suction stroke in a small perforated vaporizer extending slightly into the cylinder from the cylinder head. During the compression stroke a part of the charge is vaporized, and it is ignited at the termination of the stroke; the resulting explosion sweeps the remainder of the less readily vaporized fuel through the perforations and into the cylinder, in which it is burned completely. Gas oils are successfully burned in these engines, with a fuel consumption of 0.5 to 0.6 pound per horsepower-hour.

A further development lies in building small two-stroke engines of this type, to permit a lower weight and lower manufacturing costs. A fuel consumption of 0.6 to 0.7 pound per horsepower-hour is rather common in these engines, which range from 5 horsepower per cylinder up. What their development will mean to farmers and users of motor trucks in a time of rising gasoline prices may be realized from a comparison of fuel prices. Gas oil may be had at \$1 to \$1.50 per barrel, or 2 to 3 cents per gallon, whereas distillates and gasolines cost 9 to 25 cents per gallon. The fuel saving is therefore nearly proportional to the fuel prices, as the thermal efficiency of these small oil engines is fully as good as that of distillate and gasoline engines. Standardized engines of the type designed for mass production will reduce the manufacturing cost to such a figure as to make their adoption general throughout the country.

### EXAMPLES OF SUCCESSFUL USE OF DIESEL ENGINES.

A few practical examples of uses to which Diesel engines are adapted will show what performance may be expected. The Diesel engine is the preeminent prime mover for continuous day-and-night duty at a high load, the fuel saving in such use being large in comparison with the consumption of any other heat engine, and the operating and fixed charges for the Diesel engine so used being the lowest for the unit of power.

### PUMPING PLANTS.

The table following shows the amount of work performed, in foot-pounds per 1 B. t. u. of heat consumed, in lifting water moderate



heights with different types of pumping equipment. These performances relate to city water works and irrigation pumping plants.

*Data regarding amount of work performed by different types of pumping equipment.*

| Type of pumping plant.                             | Work performed in lifting water.  | Over-all efficiency. |
|----------------------------------------------------|-----------------------------------|----------------------|
|                                                    | <i>Foot-pounds per 1 B. t. u.</i> | <i>Per cent.</i>     |
| Steam; good operating conditions.....              | 54.5                              | 7                    |
| Steam; best operating conditions.....              | 77.8                              | 10                   |
| Steam; superheated steam used.....                 | 93.4                              | 12                   |
| Suction gas engine; good operating conditions..... | 116.7                             | 15                   |
| Suction gas engine; special conditions.....        | 147.8                             | 19                   |
| Humphrey gas pump; good operating conditions.....  | 156                               | 20.1                 |
| Humphrey gas pump; special conditions.....         | 171.2                             | 22                   |
| Diesel engine; good operating conditions.....      | 225.6                             | 29                   |

A noteworthy Diesel-engine pumping station is that at the Gladstone Docks, Liverpool. There are five units, each consisting of a single-stage centrifugal pump coupled direct to a 1,000-brake horsepower Carels Diesel engine. The engines have a two-stroke cycle, with four cylinders 510 mm. in diameter by 660 mm. stroke, and deliver 1,000 brake horsepower at 180 revolutions per minute. They can be overloaded 10 per cent. The pumps, of the Worthington type, are designed to deliver 9,200 cubic feet of water per minute against a head of 47 feet at 180 revolutions per minute, but are capable of working against a maximum head of 61 feet.

Diesel engines are also successfully used for driving pumps of petroleum pipe lines when the oil has the desired viscosity at ordinary temperatures.

#### ICE MANUFACTURE.

Developments in the manufacture of ice have made possible the utilization of Diesel engines for driving the ammonia compressors, with a noteworthy fuel saving and reduction in the cost of manufacture. Combination ice, refrigeration, and electric-light plants are now frequently installed. The off-peak engine capacity is used for operating the ice-making machinery, with highly satisfactory results. Four to 6 gallons of fuel oil per ton of ice is the usual consumption in ice plants in which Diesel engines are used. In a plant producing 100 tons of ice in 24 hours and with fuel costing  $2\frac{1}{2}$  cents a gallon, a ton of ice can be produced for 45 cents, including all operating labor, maintenance, and supply charges, but not fixed charges (interest and amortization). If the power were purchased at 1 cent per kilowatt-hour the cost would be 85 cents per ton of ice.

#### FLOUR MILLS.

A number of flour mills use Diesel engines. A common performance is the milling of one barrel of flour with a fuel-oil consumption of 0.5 to 0.6 gallon, for all power, including that used in elevating the

wheat and corn. With oil costing 2 cents a gallon, the fuel cost per barrel of flour for all power is 1 to 1.2 cents, and with 3-cent oil the cost is 1.5 to 1.8 cents per barrel of flour.

### CENTRAL STATIONS.

The factors controlling the cost of generating electric current have already been discussed. Following are data showing unit power-production costs for a modern central station in Texas. The station is equipped with three 500-horsepower Diesel engines direct connected to 437-k. v. a., 2,300-volt, 3-phase, 60-cycle alternators. The cost figures are for a period of four months.

|                                                          |             |
|----------------------------------------------------------|-------------|
| Station output, kilowatt-hours .....                     | 1, 565, 000 |
| Rating of plant, kilowatts .....                         | 1, 050      |
| Station factor, per cent. ....                           | 51          |
| Total fuel oil used, gallons.....                        | 149, 072    |
| Weight of oil per kilowatt-hour of output, pounds.....   | 0. 672      |
| Heat consumed per kilowatt-hour of output, B. t. u. .... | 13. 100     |
| Cost of plant per kilowatt of capacity, dollars.....     | 153         |
| Labor cost, average man-day, dollars.....                | 2. 80       |
| Man-days per 1,000 kilowatt-hours of output.....         | . 46        |
| Cost of fuel oil per gallon, cents.....                  | 3. 25       |
| Cost of lubricating oil per gallon, cents.....           | 30          |

#### Production costs, per kilowatt-hour, mills:

|                                                                          |       |
|--------------------------------------------------------------------------|-------|
| All labor .....                                                          | 1. 44 |
| Fuel oil .....                                                           | 3. 07 |
| Water .....                                                              | . 09  |
| Lubricants and waste .....                                               | . 04  |
| Miscellaneous supplies and expense .....                                 | . 10  |
| Maintenance of engines .....                                             | . 04  |
| Maintenance of buildings .....                                           | . 05  |
| All other maintenance.....                                               | . 15  |
| Total.....                                                               | 4. 98 |
| Interest and amortization, 12 per cent, 51 per cent station factor ..... | 4. 80 |
| Total cost.....                                                          | 9. 78 |

### LOCOMOTIVES.

The Diesel engine has been applied to railroad traction in two forms; directly in a Diesel locomotive, and indirectly in an electric motor car with a generator driven by a Diesel engine.

A 1,000-horsepower locomotive has been built by Sulzer Bros., of Winterthur, Switzerland, in which power to the drivers is transmitted direct from a reversible Diesel engine with a two-stroke cycle. For starting, compressed air, furnished by a compressor operated by a separate Diesel engine is used. The locomotive made satisfactory test runs in hauling freight and passenger trains, but no detailed performance data are as yet available.

Variable-speed Diesel engines coupled to generators and built into passenger railroad cars have been used in several instances in Europe with great success. The fuel consumption varied from 0.8 pound to 1.2 pounds per train-mile, with a train weight (two cars) of 30 to 66 tons (2,000 pounds). The average fuel consumption amounted to 2 to 2.75 pounds per 100 ton-miles. On a basis of 1 kilowatt-hour being generated with a fuel consumption of 0.7 pound of oil, this is equal, in round numbers, to 3 to 4 kilowatt-hours per 100 ton-miles, or 30 to 40 watt-hours per ton-mile, a remarkably good performance.

### SHIP PROPULSION.

An increasing number of ships are being fitted with Diesel engines. Lloyd's Register for 1914 listed 27 ocean-going ships equipped with Diesel engines with a total of about 40,000 shaft horsepower. At least 20 more ships equipped with Diesel engines were then building. There are probably more than 400 ships, including river craft not listed, fitted with Diesel engines, not counting submarines so fitted. Up to 1914, 6,000 indicated horsepower, equal to about 4,600 to 4,800 shaft horsepower, in six single-acting cylinders and in one engine was probably the upper limit in size, although double-acting engines of 12,000 horsepower in three cylinders for use on ships are being tested on the European continent.

Before directly reversible marine Diesel engines had been developed, reversing and slow-speed changes were accomplished by the Del Proposto system, first used on the Russian ship *Sarmat* in 1904, and subsequently installed in many Russian river boats and ships of the Russian Navy. The engine was used for driving the ship ahead, but for reversing was coupled to a generator and a motor was used. The engine speed could be reduced from 240 to 72 revolutions per minute.

Later, Hesselmann developed a directly reversible four-cycle marine engine, which has been used with great success in many ocean-going ships. Late development favors the engine with a two-stroke cycle for marine purposes, as it can be built materially lighter per horsepower, is mechanically better adapted for reversing, and occupies much less space than other types. Compressed air is used for reversing both types of engines.

The advantages of ships equipped with Diesel engines lie not merely in the greater cruising radius obtainable with a given quantity of fuel, but in the increased cargo space made available, as the liquid fuel can be stored between the double bottoms of the ships, in dead space otherwise filled with ballast water.

The importance of Diesel engines for propelling submarines is commonly known. The latest types of submarines with their greatly increased field of action and cruising radius are almost wholly due to recent improvements in the marine Diesel engine.



## SELECTED BIBLIOGRAPHY.

1893.

- DIESEL, RUDOLF. Theorie und Konstruktion eines rationellen Wärmemotors zum Ersatz der Dampfmaschinen und der heute bekannten Verbrennungsmotoren. Berlin, 1893, 96 pp.
- KÖHLER, OTTO. Der rationelle Wärmemotor in Vergleich mit andern Wärmemotoren. Ztschr. Ver. deut. Ing., Bd. 37, Sept. 9, 1893, pp. 1103-1109.

1897.

- DIESEL, RUDOLF. Diesels rationeller Wärmemotor. Ztschr. Ver. deut. Ing., Bd. 41, July 10, 17, 1897, pp. 786-791, 817-821.
- MEYER, E. Die Beurteilung der Kreisprozesse von Wärmekraftmaschinen mit besonderer Berücksichtigung des Diesel Motors. Ztschr. Ver. deut. Ing., Bd. 41, Sept. 25, 1897, pp. 1108-114.
- SCHRÖTER, M. Diesels rationeller Wärmemotor. Ztschr. Ver. deut. Ing., Bd. 41, July 24, 1897, pp. 845-852.

1898.

- BÁNKI, DONÁT. Zur Theorie der Wärmemotoren. Ztschr. Ver. deut. Ing., Bd. 42, Aug. 13, 1898, pp. 893-902.

1899.

- DIESEL, RUDOLF. Mitteilungen über den Dieselschen Wärmemotor. Ztschr. Ver. deut. Ing., Bd. 43, Jan. 14, 1899, pp. 36-42, 128-130.

1906.

- DUBBEL, H. Kraftmaschinen auf der Bayerischen Landesausstellung in Nürnberg. Ztschr. Ver. deut. Ing., Bd. 50, Nov. 3, 1906, pp. 1788-1796.
- MEUTH, H. Die Wärmekraftmaschinen der Jubiläums-Landesausstellung in Nürnberg. 1906. Ding. Polytech. Jour., Bd. 321, Dec. 29, 1906, pp. 818-822

1907.

- KUTZBACH, KARL. Die flüssigen Brennstoffe und ihre Ausnutzung in der Verbrennungskraftmaschine, mit besonderer Berücksichtigung des Dieselmotors. Ztschr. Ver. deut. Ing., Bd. 51, Apr. 6, 1907, pp. 521-527.
- RIEPPEL, PAUL. Versuche über die Verwendung von Teerölen zum Betrieb des Dieselmotors. Ztschr. Ver. deut. Ing., Bd. 51, Apr. 20, 1907, pp. 613-618.

1908.

- DINGLERS POLYTECHNISCHES JOURNAL. Versuche an einem Dieselmotor der Gasmotoren-Fabrik Deutz. Bd. 323, Feb. 8, 1908, pp. 95-96.
- EBERLE, C. Versuche an einem raschlaufenden Dieselmotor. Ztschr. Ver. deut. Ing., Bd. 52, Feb. 1, 1908, pp. 178-182.

1909.

- HOELTJE, E. Der Brennstoffverbrauch von modernen Heissdampflokomoiblen und Dieselmotoren im praktischen Betriebe. Ztschr. Ver. deut. Ing., Bd. 53, May 15, 1909, pp. 784-786.
- LUNGE, GEORG. Coal-tar and ammonia. Lond., 1909, 2 vols.
- MURAUER, R. Die Entwicklung und Verbreitung des Dieselmotors in Russland und seine Verwendung als Schiffsmotor. Ztschr. Ver. deut. Ing., Bd. 53, July 24, 31, 1909, pp. 1184-1188, 1236-1239.

## 1910.

- DINGLERS POLYTECHNISCHES JOURNAL. Werkstättenbetriebe mit Sulzer-Dieselmotoren. Bd. 325, Mar. 26, 1910, pp. 187-190.
- KUSTER, J. Die Motoren auf der internationalen Motorbootausstellung Berlin 1910. Ding. Polytech. Jour., Bd. 325, June 25 and June 26, 1910, pp. 385-387.
- MATHOT, R. E. The construction and working of internal-combustion engines. New York, 1910, 554 pp.
- ROMBERG, F. Leistungs und Verbrauchsversuche an einem schnellaufenden Rohöl-Kleinmotor von 5 PS. Ztschr. Ver. deut. Ing., Bd. 54, Nov. 5, 1910, pp. 1897-1901.

## 1911.

- DIESEL, RUDOLF. Ueberblick über den heutigen Stand des Dieselmotorbaues und die Versorgung mit flüssigen Brennstoffen. Ztschr. Ver. deut. Ing., Bd. 55, Aug. 12, 1911, pp. 1345-1351.
- DUBBEL, H. Der Kraftmaschinenbau auf der Weltausstellung in Brüssel. Ztschr. Ver. deut. Ing., Bd. 55, Feb. 4, 1911, pp. 166-172.
- KAEMMERER, W. Die internationale Industrie- und Gewerbeausstellung in Turin 1911. Ztschr. Ver. deut. Ing., Bd. 55, July, 1911, pp. 1244-1251.
- LUNGE, GEORG. Technical method of chemical analysis. (English edition of Lunge's "Chemisch-technische Untersuchungs Methoden.") Chapter on coal tars by Dr. H. Köhler, in vol. 2, pt. 2, p. 751.
- MILTON, J. T. Diesel engines for sea-going vessels. Engineering, vol. 91, Apr. 14, 1911, pp. 472-477, 495-499.
- NÄGEL, A. Die neuere Entwicklung der ortfesten Ölmaschine. Ztschr. Ver. deut. Ing., Bd. 55, Aug. 12, 1911, pp. 1318-45.
- HOTTINGER, M. Die Abwärmeausnutzung bei Dieselmotoren. Ztschr. Ver. deut. Ing., Bd. 55, Apr. 29, 1911, pp. 673-678.
- PÖHLMANN, C. Neuere Rohölmotoren. Ding. Polytech. Jour., Bd. 326, July 8, 22; Oct. 7, 14, 21, 28; Nov. 4; and Dec. 16, 1911, pp. 421-422, 453-455, 625-631, 640-646, 673-676, 689-693, 785-787.
- SCHUBELER, J. F. Modern Diesel oil-engines. Engineering, vol. 92, July 28, 1911, pp. 148-149; Proc. Inst. Mech. Eng., July, 1911, pp. 579-602.
- SEILIGER, M. Thermodynamische Untersuchung schnellaufender Dieselmotoren. Ztschr. Ver. deut. Ing., Bd. 55, Apr. 15, 22, 1911, pp. 587-592, 625-628.
- ZEITSCHRIFT DES VEREINES DEUTSCHER INGENIEURE. Verwendung von Vertikal-ofen-Teer für Dieselmotoren. Bd. 55, Apr. 22, 1911, pp. 649-650.

## 1912.

- COCHAND, J., AND HOTTINGER, M. Versuche an einer Sulzerschen 300 pferdigen Dieselmotorenanlage mit Abwärmeverwertung. Ztschr. Ver. deut. Ing., Bd. 56, Mar. 23, 1912, pp. 458-463.
- ENGINEERING. Commercial prospects of the marine oil engine. Vol. 94, Nov. 1, 1912, pp. 609-610.
- The Diesel engine. Vol. 94, Dec. 13, 1912, pp. 815-816.
- The Diesel-engined-ship *Fordonian*. Vol. 94, Sept. 27, 1912, pp. 435-436.
- Four-cylinder four-cycle Diesel engines. Vol. 93, May 24, 1912, pp. 696-699.
- The Italian tugboat *Savoia* with Diesel engines. Vol. 94, Dec. 6, 1912, p. 777.
- The oil-engined ship *Fordonian*. Vol. 94, Dec. 13, 1912, pp. 807-808.
- Oil-tank steamer driven by Diesel engines. Vol. 93, Apr. 19, 1912, pp. 526-527.
- HOLDE, D. Untersuchung der Mineralöle und Fette. 3d ed., Berlin, 1912, 459 pp.
- KNUDSEN, I. Results of trials of the Diesel-engined sea-going vessel *Selandia*. Engineering, vol. 93, Apr. 5, 1912, pp. 439-440, 448-450.
- LANE, F. Fuels for Diesel engines. Engineering, vol. 94, Oct. 4, 1912, pp. 471-472.

- LUNGE, GEORG, AND KÖHLER, HIPPOLYT. Die Industrie des Steinkohlenteers und des Ammoniaks. Braunschweig, 1912, Bd. 1 and 2.
- ORDE, E. L. Diesel oil engine. Engineering, vol. 93, May 24, 1912, pp. 709-710.
- PÖHLMANN, C. Neuere Rohölmotoren. Ding. Polytech. Jour., Bd., 327, Jan. 27; Feb. 10, 17, 24; Mar. 16, and 30, 1912, pp. 49-51, 81-87, 97-102, 119-121, 165-167, 193-197, 229-232.
- SHANNON, D. M. Some aspects of Diesel engine design. Engineering, vol. 93, May 3, 1912, pp. 605-610.

## 1913.

- ENGINEERING. The cargo ship *France*. Vol. 96, Oct. 10, 1913, pp. 491-494.
- Diesel engine cylinder dimensions. Vol. 96, Sept. 26, 1913, pp. 430-431.
- Diesel engine pumping-station at the Gladstone dock, Liverpool. Vol. 96, Sept. 12, 1913, pp. 349-352.
- Diesel marine oil engines. Vol. 96, Dec. 19, 1913, pp. 817-819.
- Internal-combustion engines at the Ghent exhibition. Vol. 96, Aug. 1, 1913, pp. 161-162.
- 100-brake-horsepower Junkers oil engine. Vol. 96, Oct. 24, 1913, p. 567.
- The performance of the Diesel-engined ship *Monte Ponedo*. Vol. 96, Sept. 12, 1913, p. 364.
- Reavell air compressors for Diesel engines. Vol. 95, June 27, 1913, p. 864.
- Results of first voyage of the oil-engined ship *Christian X*. Vol. 95, Jan. 3, 1913, pp. 22-23.
- The Sulzer Diesel locomotive. Vol. 96, Sept. 5, 1913, pp. 317-319.
- GÜLDNER, HUGO. Das entwerfen und Berechnen der Verbrennungsmotoren und Kraftgas-Anlagen. Berlin, 1913.
- HAAS, HERBERT. The principles of the Diesel oil engine. Eng. and Min. Jour., Apr. 26, 1913, pp. 843-848.
- KNUDSEN, I. Performance in the service of the motor-ship *Suecia*. Engineering, vol. 96, July 4, 1913, p. 27.
- LÜDERS, JOHANNES. Der Dieselmithus; quellenmässige Geschichte der Entstehung des heutigen Ölmotors. Berlin, 1913, 236 pp.
- MEYER, P. Die Verbrennungskraftmaschinen auf der Weltausstellung in Ghent 1913. Ztschr. Ver. deut. Ing., Bd. 57, Sept., 1913, pp. 1463-1466.
- REAVELL, W. Compressed air for working auxiliaries in ships propelled by internal-combustion engines. Engineering, vol. 95, Mar. 21, 1913, pp. 403-405.
- SIMON, R. Einlassventil für Teeröl-Dieselmotoren. Ding. Polytech. Jour., Bd. 328, July 26, 1913, pp. 466-469.
- STERNENBERG, F. Die erste Thermo-Lokomotive. Ztschr. Ver. deut. Ing., Bd. 57, Aug. 23, 1913, pp. 1325-1331.
- SUPINO, GIORGIO. Dieselmotoren. Trans. by H. Zeman, Munich and Berlin, 1913, 233 pp.
- ZEITSCHRIFT DES VEREINES DEUTSCHER INGENIEURE. Grossdieselmotoren, ihre Brennstoffe, Konstruktion und Anwendungsgebiete. Bd. 57, July 19, 1913, pp. 1144-1148.
- ZERKOWITZ, G. Über die Beurteilung der Wärmekraftmaschinen. Ding. Polytech. Jour., Bd. 328, Nov. 29, 1913, pp. 755-758.

## 1914.

- BARTH, FRIEDRICH. Liegende doppeltwirkende Viertakt-Dieselmotoren. Ztschr. Ver. deut. Ing., Bd. 58, Aug. 1, 1914, pp. 1242-1250.
- BURNS, W. S. Some tests on a Diesel engine. Engineering, vol. 98, Aug. 21, 1914, pp. 263-266.
- CONSTAM, E. J., AND SCHLÄFFER, P. Ueber Treiböle. Ztschr. Ver. deut. Ing., Bd. 57, Sept. 20, 1913, pp. 1489-1493.



- DALBY, W. E. Trials of a small Diesel engine. *Engineering*, vol. 97, Apr. 10, 1914, pp. 504-509.
- DROSNE, M. Development of internal-combustion engines for marine purposes. *Proc. Inst. Mech. Eng.*, July, 1914, pp. 537-557.
- ENGINEERING. Carels' marine oil engines. Vol. 97, Mar. 20, 1914, pp. 377-379.
- Marine Diesel oil engine (Sulzer type). Vol. 97, Jan. 2, 1914, pp. 15-16.
- The motor ship *Fionia*. Vol. 97, Jan. 9, 1914, pp. 44-45.
- Oil-engine propelled ships of the East Asiatic Co. Vol. 97, Mar. 13, 1914, p. 360.
- Three-masted schooner *Aosta* with Savoia Diesel marine engines. Vol. 97, Feb. 6, 1914, pp. 182-183.
- FÖTTINGER, H. Recent development of the hydraulic transformer. *Engineering*, vol. 98, Sept. 25, 1914, pp. 397-401.
- FULLAGAR, H. F. Balancing of internal-combustion engines. *Engineering*, vol. 98, July 17, 1914, pp. 103-105.
- HOFMANN, MAX. Berkmerkungen über doppeltwirkende Zweitaktmotoren. *Ding. Polytech. Jour.*, Bd. 329, Sept. 26, 1914, pp. 575-577.
- HOPKINSON, B. The charging of two-cycle internal-combustion engines. *Engineering*, vol. 98, July 17, 1914, pp. 100-102.
- KREUL, WILHELM. Neuere doppeltwirkende Zweitakt-Dieselmotoren. *Ding. Polytech. Jour.*, Bd. 329, Mar. 12, Apr. 4, 11, 1914, pp. 177-179, 215-218, 231-234.
- MEUTH, H. Die Oelmaschine System Junkers. Bd. 329, Jan. 3, 1914, pp. 1-4.
- MILTON, J. T. The marine Diesel engine. *Engineering*, vol. 97, Apr. 10, 1914, pp. 501-504.
- MÜNZINGER, FRIEDRICH. Untersuchungen an einem 15 pferdigen M. A. N-Dieselmotor. *Ztschr. Ver. deut. Ing.*, Bd. 58, June 27, 1914, pp. 1049-1056.
- RICHARDSON, JAMES. The development of high power marine Diesel engines. *Engineering*, vol. 97, Apr. 24, 1914, pp. 573-578.
- SCHRAUFF, —. Der Bau von Dieselmotoren in den Vereinigten Staaten von Amerika. *Ztschr. Ver. deut. Ing.*, Bd. 58, July 25, 1914, pp. 1201-1207.
- SETZ, H. R. Recent developments in the manufacture of the Diesel engine. *Jour. Am. Soc. Mech. Eng.*, vol. 36, 1914, pp. 420-434.
- WOLFF, H. Berechnung der Kanallängen von Zweitakt-Oelmaschinen mit Schlitzsteuerung. *Ding. Polytech. Jour.*, Bd. 329, Apr. 25, 1914, pp. 264-265.

## 1915.

- ADAMS, W. H. The Diesel engine and its application in Southern California. *Trans. Am. Soc. Mech. Eng.*, vol. 37, 1915, pp. 447-476.
- CHALKLEY, A. P. Diesel engines for land and marine work. Lond., 1915, 368 pp.
- ENGINEERING. The Burmeister and Wain oil engine. Vol. 99, Feb. 19, 1915, p. 209.
- Machinery of Western-Australian Government motor-ship *Kangaroo*. Vol. 100, Dec. 3, 1915, pp. 565-568.
- GOLDINGHAM, A. H. The heavy oil engine, its present status and future development. *Trans. Am. Soc. Mech. Eng.*, vol. 37, 1915, pp. 477-501.
- HAAS, HERBERT. Diesel engine installation at Palo Alto, California. *Power*, vol. 41, Apr. 13, 1915, pp. 502-504.
- LEES, S. The effect of variation of specific heat with temperature on the theoretical efficiency of the Diesel engine. *Engineering*, vol. 99, Jan. 1, 1915, pp. 1-2.
- LOW, F. R. Diesel engines at the Panama-Pacific Exposition. *Power*, vol. 42, Aug. 24, 1915, pp. 261-290.
- McIntosh and Seymour 500-H. P. Diesel engine. *Power*, vol. 42, Nov. 30, 1915, pp. 740-744.
- POWER. The Diesel principle applied to small engines. Vol. 41, Feb. 2, 1915, pp. 162-166.
- The Southwark-Harris Diesel engine. Vol. 41, June 29, 1915.

- ROTTER, MAX. The Diesel engine in America. Trans. Int. Eng. Cong., 1915, Mechanical Engineering. 1915, pp. 296-330.
- SUPINO, GIORGIO. Land and marine Diesel engines; trans. by A. G. Bremmer and J. Richardson. Lond., 1915, 309 pp.
- TAKEMURA, KANGO. Theoretical efficiency of the Diesel constant-temperature cycle, taking the specific heat to be a linear function of the temperature. Engineering, vol. 100, Nov. 26, 1915, pp. 540-541, 568.
- WATSON, H. L. Internal-combustion engine dimensions. Power, vol. 41, May 18, 1915, pp. 672-674.
- WELLS, G. J., AND WALLIS-TAYLOR, A. J. The Diesel or slow-combustion oil engine. Lond., 1915, 304 pp.
- 1916.
- ENGINEERING. The 4,600-B. H. P. marine Diesel engines of the motor vessel *Ceara*. Vol. 101, Feb. 25, 1916, pp. 180-181.
- The resistance to wear of cast-iron liners and cylinders. Vol. 101, Feb. 18, 1916, pp. 149-150.
- The twin-screw motor ship *Abelia*. Vol. 102, Oct. 20 and Nov. 3, 1916, pp. 379, 434.
- The twin-screw motor ship *Peru*. Vol. 102, Oct. 6, 1916, pp. 324-325.
- Use of tar and tar oils in Diesel engines. Vol. 101, Apr. 7, 1916, p. 339.
- FORD, J. M. High-pressure air compressors. Engineering, vol. 102, Nov. 17, 1916, pp. 480-481.
- HAAS, HERBERT. Diesel engines versus steam turbines for mine power plants. Trans. Am. Inst. Min. Eng., vol. 55, Sept., 1916, pp. 161-181.
- HURST, J. E. The growth of internal-combustion engine cylinders. Engineering, vol. 102, Aug. 4, 1916, pp. 97-98.
- MATHOT, R. E. Margin of power in internal-combustion engines. Power, vol. 43, June 20, 1916, pp. 877-878.
- MOORE, H. H. Fuel oils from coal. Engineering, vol. 101, Mar. 3, 1916, p. 206.
- PORTER, GEOFFREY. Oil engines and steam engines in combination. Engineering, vol. 102, July 21, 1916, pp. 68-70.
- POWER. Cooling system for Diesel engine circulating water. Vol. 43, Jan. 18, 1916, pp. 77-78.
- Diesel engine operation at Palo Alto municipal plant. Vol. 43, May 2, 1916, pp. 610-611.
- SILLINCE, W. P. The marine Diesel engine and its possibilities. Engineering, vol. 101, Apr. 21, 1916, pp. 388-390.
- STEINHEIL, G. Trial of a 600-B. H. P. two-stroke direct reversible Noble Diesel engine. Engineering, vol. 102, Dec. 22, 1916, pp. 608-612.
- WILKINS, F. T. Diesel engine trials. Engineering, vol. 102, Oct. 27, 1916, pp. 422-425.

## PUBLICATIONS ON PETROLEUM TECHNOLOGY.

A limited supply of the following publications of the Bureau of Mines has been printed and is available for free distribution until the edition is exhausted. Requests for all publications can not be granted, and to insure equitable distribution applicants are requested to limit their selection to publications that may be of especial interest to them. Requests for publications should be addressed to the Director, Bureau of Mines.

The Bureau of Mines issues a list showing all its publications available for free distribution, as well as those obtainable only from the Superintendent of Documents, Government Printing Office, on payment of the price of printing. Interested persons should apply to the Director, Bureau of Mines, for a copy of the latest list.

### PUBLICATIONS AVAILABLE FOR FREE DISTRIBUTION.

BULLETIN 120. Extraction of gasoline from natural gas by absorption methods, by G. A. Burrell, P. M. Biddison, and G. G. Oberfell. 1917. 71 pp., 2 pls., 15 figs.

BULLETIN 134. The use of mud-laden fluid in oil and gas wells, by J. O. Lewis and W. F. McMurray. 1916. 86 pp., 3 pls., 18 figs.

BULLETIN 148. Methods for increasing the recovery of oil from wells, by J. O. Lewis. 1917. 28 pp., 4 pls., 32 figs.

TECHNICAL PAPER 32. The cementing process of excluding water from oil wells as practiced in California, by Ralph Arnold and V. R. Garfias. 1913. 12 pp., 1 fig.

TECHNICAL PAPER 37. Heavy oil as fuel for internal-combustion engines, by I. C. Allen. 1913. 36 pp.

TECHNICAL PAPER 38. Wastes in the production and utilization of natural gas, and methods for their prevention, by Ralph Arnold and F. G. Clapp. 1913. 29 pp.

TECHNICAL PAPER 42. The prevention of waste of oil and gas from flowing wells in California, with a discussion of special methods used by J. A. Pollard, by Ralph Arnold and V. R. Garfias. 1913. 15 pp., 2 pls., 4 figs.

TECHNICAL PAPER 45. Waste of oil and gas in the Mid-Continent fields, by R. S. Blatchley. 1914. 54 pp., 2 pls., 15 figs.

TECHNICAL PAPER 66. Mud-laden fluid applied to well drilling, by J. A. Pollard and A. G. Heggem. 1914. 21 pp., 12 figs.

TECHNICAL PAPER 68. Drilling wells in Oklahoma by the mud-laden fluid method, by A. G. Heggem and J. A. Pollard. 1914. 27 pp., 5 figs.

TECHNICAL PAPER 70. Methods of oil recovery in California, by Ralph Arnold and V. R. Garfias. 1914. 57 pp., 7 figs.

TECHNICAL PAPER 72. Problems of the petroleum industry, results of conferences at Pittsburgh, Pa., August 1 and September 10, 1913, by I. C. Allen. 1914. 20 pp.

TECHNICAL PAPER 79. Electric lights for oil and gas wells, by H. H. Clark. 1914. 8 pp.

TECHNICAL PAPER 87. Methods of testing natural gas for gasoline content, by G. A. Burrell and G. W. Jones. 1916. 26 pp., 7 figs.



TECHNICAL PAPER 104. Analysis of natural gas and illuminating gas by fractional distillation in a vacuum at low temperatures and pressures, by G. A. Burrell, F. M. Seibert, and I. W. Robertson. 1915. 41 pp., 7 figs.

TECHNICAL PAPER 117. Quantity of gasoline necessary to produce explosive vapors in sewers, by G. A. Burrell and H. T. Boyd. 1916. 18 pp., 4 figs.

TECHNICAL PAPER 127. Hazards in handling gasoline, by G. A. Burrell. 1915. 12 pp.

TECHNICAL PAPER 130. Underground wastes in oil and gas fields and methods of prevention, by W. F. McMurray and J. O. Lewis. 1916. 28 pp., 1 pl., 8 figs.

TECHNICAL PAPER 131. The compressibility of natural gas at high pressures, by G. A. Burrell and I. W. Robertson. 1916. 11 pp., 2 figs.

TECHNICAL PAPER 161. Construction and operation of a single-tube cracking furnace for making gasoline, by C. P. Bowie. 1916. 16 pp., 10 pls.

TECHNICAL PAPER 163. Physical and chemical properties of gasolines sold throughout the United States during the calendar year 1915, by W. F. Rittman, W. A. Jacobs, and E. W. Dean. 1916. 45 pp., 4 figs.

#### PUBLICATIONS THAT MAY BE OBTAINED ONLY THROUGH THE SUPERINTENDENT OF DOCUMENT.

BULLETIN 19. Physical and chemical properties of the petroleum of the San Joaquin Valley, Cal., by I. C. Allen and W. A. Jacobs, with a chapter on analyses of natural gas from the southern California oil fields, by G. A. Burrell. 1911. 60 pp., 2 pls., 10 figs. 10 cents.

BULLETIN 32. Commercial deductions from comparisons of gasoline and alcohol tests on internal-combustion engines, by R. M. Strong. 36 pp. 5 cents.

BULLETIN 43. Comparative fuel values of gasoline and denatured alcohol in internal-combustion engines, by R. M. Strong and Lauson Stone. 1912. 243 pp., 3 pls., 32 figs. 20 cents.

BULLETIN 65. Oil and gas wells through workable coal beds; papers and discussions, by G. S. Rice, O. P. Hood, and others. 1913. 101 pp., 1 pl., 11 figs. 10 cents.

BULLETIN 88. The condensation of gasoline from natural gas, by G. A. Burrell, F. M. Seibert, and G. G. Oberfell. 1915. 106 pp., 6 pls., 18 figs. 15 cents.

BULLETIN 114. Manufacture of gasoline and benzene-toluene from petroleum and other hydrocarbons, by W. F. Rittman, C. B. Dutton, and E. W. Dean, with a bibliography compiled by M. S. Howard. 1916. 263 pp., 9 pls., 45 figs. 35 cents.

BULLETIN 125. The analytical distillation of petroleum, by W. F. Rittman and E. W. Dean. 1916. 79 pp., 1 pl., 16 figs. 15 cents.

TECHNICAL PAPER 3. Specifications for the purchase of fuel oil for the Government, with directions for sampling oil and natural gas, by I. C. Allen. 1911. 13 pp. 5 cents.

TECHNICAL PAPER 10. Liquefied products of natural gas, their properties and uses, by I. C. Allen and G. A. Burrell. 1912. 23 pp. 5 cents.

TECHNICAL PAPER 25. Methods for the determination of water in petroleum and its products, by I. C. Allen and W. A. Jacobs. 1912. 13 pp., 2 figs. 5 cents.

TECHNICAL PAPER 26. Methods for the determination of the sulphur content of fuels, especially petroleum products, by I. C. Allen and I. W. Robertson. 1912. 13 pp., 1 fig. 5 cents.

TECHNICAL PAPER 36. The preparation of specifications for petroleum products, by I. C. Allen. 1913. 12 pp. 5 cents.

TECHNICAL PAPER 51. Possible causes of the decline of oil wells and suggested methods of prolonging yield, by L. G. Huntley. 1913. 32 pp., 9 figs. 5 cents.

TECHNICAL PAPER 53. Proposed regulations for the drilling of oil and gas wells, with comments thereon, by O. P. Hood and A. G. Heggem. 1913. 28 pp., 2 figs. 5 cents.

TECHNICAL PAPER 57. A preliminary report on the utilization of petroleum and natural gas in Wyoming, by W. R. Calvert, with a discussion of the suitability of natural gas for making gasoline, by G. A. Burrell. 1913. 23 pp. 5 cents.

TECHNICAL PAPER 74. Physical and chemical properties of the petroleums of California, by I. C. Allen, W. A. Jacobs, A. S. Crossfield, and R. R. Matthews. 1914. 38 pp., 1 fig. 5 cents.

TECHNICAL PAPER 109. Composition of the natural gas used in 25 cities, with a discussion of the properties of natural gas, by G. A. Burrell and G. G. Oberfell. 1915. 22 pp. 5 cents.

TECHNICAL PAPER 115. Inflammability of mixtures of gasoline vapor and air, by G. A. Burrell and H. T. Boyd. 1915. 18 pp., 2 figs. 5 cents.

TECHNICAL PAPER 120. A bibliography of the chemistry of gas manufacture, by W. F. Rittman and N. C. Whittaker; compiled and arranged by M. S. Howard. 1915. 30 pp. 5 cents.

## INDEX

|                                                                             | A. | Page.   |
|-----------------------------------------------------------------------------|----|---------|
| Air for starting Diesel engine, pressure of....                             |    | 8       |
| Air-inlet valve, position of.....                                           |    | 35      |
| Air ports, position of.....                                                 |    | 35      |
| Allis-Chalmers Manufacturing Co., Diesel engine of.....                     |    | 70      |
| figure showing.....                                                         |    | 70, 71  |
| Anthracite, average heat cost of.....                                       |    | 98      |
| Arachis oil, as Diesel engine fuel.....                                     |    | 93      |
| Asphalt content of fuel oil, definition of.....                             |    | 83      |
| Ash content of fuel oil, objections to.....                                 |    | 81      |
| permissible, of fuel oil.....                                               |    | 88      |
| Atomization of fuel, importance of.....                                     |    | 38      |
| Atomizer, common type of.....                                               |    | 38-41   |
| figure showing.....                                                         |    | 39      |
| B.                                                                          |    |         |
| Bearing oil, requirements of.....                                           |    | 94      |
| Bearings, main, lubrication of, figure showing,                             |    | 24      |
| Benzol, combustion of.....                                                  |    | 78      |
| unsuitability of for engine fuel.....                                       |    | 78      |
| "Benzol ring," description of.....                                          |    | 78      |
| Bibliography.....                                                           |    | 123-127 |
| Bituminous coal, average heat cost of.....                                  |    | 98      |
| Blast-furnace gas, average heat cost of.....                                |    | 93      |
| Boiling point of fuel for Diesel engine.....                                |    | 87      |
| Boiling-point tests of petroleum, value of.....                             |    | 80      |
| Bureau of Mines, work of.....                                               |    | 1       |
| Burning point of fuel, significance of.....                                 |    | 84      |
| Busch-Sulzer Bros. Diesel Engine Co., Diesel engine of, figure showing..... |    | 70      |
| C.                                                                          |    |         |
| California crude oil, as Diesel engine fuel.....                            |    | 94      |
| Carburetion, nature of.....                                                 |    | 2       |
| Cartnot cycle, diagram showing.....                                         |    | 18      |
| Central stations, Diesel engines for.....                                   |    | 121     |
| Coal tar, classification of.....                                            |    | 91      |
| coke residue of.....                                                        |    | 92      |
| suitability of for engine fuel.....                                         |    | 92      |
| Coal tar oils, composition of.....                                          |    | 89      |
| use of in Germany.....                                                      |    | 89      |
| Coke content of fuel, effects of.....                                       |    | 82      |
| upper limit of.....                                                         |    | 80      |
| Coke-oven gas, average heat cost of.....                                    |    | 98      |
| Combustion of fuel, factors affecting.....                                  |    | 85      |
| Combustion space, construction of.....                                      |    | 86      |
| Compression pressure, in Diesel engine.....                                 |    | 6       |
| in explosion-oil engines.....                                               |    | 5       |
| See also Diesel engine.                                                     |    |         |
| Constant-pressure oil engine, definition of.....                            |    | 2       |
| Cooling water. See Water for cooling.                                       |    |         |
| Crank-effort diagrams, figure showing.....                                  |    | 30      |
| Crank shafts, lubrication of.....                                           |    | 22      |
| of Diesel engines, construction of.....                                     |    | 22      |
| Cylinder frames, advantages of.....                                         |    | 20      |
| head, construction of.....                                                  |    | 31      |
| figure showing.....                                                         |    | 30      |

|                                                       | Page.      |
|-------------------------------------------------------|------------|
| Cylinder oil, for Diesel engine, requirements of..... | 95, 96     |
| need of care in selecting.....                        | 95         |
| Cylinders, cooling of, water for.....                 | 32         |
| Diesel engine, construction of.....                   | 21         |
| lubrication of.....                                   | 28         |
| <i>See also Diesel engine.</i>                        |            |
| D.                                                    |            |
| Density of fuel oil.....                              | 85         |
| Diesel, Rudolph, cited.....                           | 10         |
| first engines of.....                                 | 18         |
| Diesel engine, advantage of.....                      | 1, 95, 98  |
| summary of.....                                       | 113        |
| available fuels for.....                              | 16         |
| commercial limits in size of.....                     | 14         |
| comparative efficiency of, figure showing.....        | 115        |
| compression-pressure in.....                          | 6          |
| cost of installation of.....                          | 17         |
| cost of lubricant for.....                            | 17         |
| cost of fuel for.....                                 | 17         |
| definition of.....                                    | 2          |
| details of construction of.....                       | 22-63      |
| efficiency of.....                                    | 16, 19, 98 |
| diagram showing.....                                  | 68         |
| examples of successful use of.....                    | 119-122    |
| for pumping at Gladstone docks, Liverpool.....        | 120        |
| four-stroke cycle, description of.....                | 6          |
| figure showing.....                                   | 7          |
| fuel consumption of, at varying loads.....            | 16         |
| fuel economy of.....                                  | 95         |
| fuels for, cheapness of.....                          | 119        |
| fuel-oil pump for, working pressure of.....           | 6          |
| fuel-supply requirements of.....                      | 96         |
| high speed, characteristics of.....                   | 69         |
| horizontal, details of.....                           | 22         |
| reliability of.....                                   | 97         |
| limiting diameter of cylinders of.....                | 14         |
| low speed, characteristics of.....                    | 69         |
| marine, improvements in.....                          | 122        |
| operating cost of.....                                | 98         |
| over-all economy of.....                              | 15         |
| power losses, figure showing.....                     | 66         |
| indicator diagram of, figure showing.....             | 13         |
| single-acting, figure showing.....                    | 10         |
| indicator diagram of, figure showing.....             | 11         |
| small, development of.....                            | 119        |
| space required by.....                                | 96         |
| starting of.....                                      | 8          |
| temperatures in.....                                  | 15         |
| thermal efficiency of.....                            | 66         |
| types of, efficiency of.....                          | 65         |
| two-stroke cycle, description of.....                 | 8          |
| figure showing.....                                   | 9          |
| single-acting, figure showing.....                    | 10         |
| indicator diagram of, figure showing.....             | 10         |



|                                                                         | Page.        |                                                          | Page. |
|-------------------------------------------------------------------------|--------------|----------------------------------------------------------|-------|
| Diesel engine, vertical, details of.....                                | 19-21        | Fuel, relative prices of.....                            | 98    |
| volumetric efficiency of.....                                           | 67           | Fuel for Diesel engines, characteristics of....          | 77    |
| <i>See also</i> Fuel oil, fuels, and firms named.                       |              | classes of.....                                          | 77    |
| Diesel engine and auxiliaries, figure showing.....                      | 99, 100, 101 | Fuel oil, ash content of, objection to.....              | 81    |
| Diesel engine and steam turbine, combinations of.....                   | 116          | asphalt content of, definition of.....                   | 83    |
| especial uses of.....                                                   | 117          | average heat cost of.....                                | 98    |
| costs of power with, figure showing.....                                | 114          | burning point of, significance of.....                   | 84    |
| Diesel and explosion oil engines, comparative economies of.....         | 17           | coke contents of, effect of.....                         | 82    |
| air for starting, pressure of.....                                      | 8            | cracking of, objection to.....                           | 80    |
| Dow Pump & Diesel Engine Co., Diesel engine of.....                     | 72           | density of, significance of.....                         | 85    |
| E.                                                                      |              | flash point of.....                                      | 84    |
| Efficiency, engine, effect of altitude on.....                          | 69           | freedom from acids of.....                               | 84    |
| mechanical.....                                                         | 65           | heavy, burning of, method of.....                        | 36    |
| thermal.....                                                            | 65           | high hydrogen content of, advantage of....               | 84    |
| volumetric.....                                                         | 67           | impurities in, removal of.....                           | 81    |
| Efficiencies, engine, figure showing.....                               | 68           | heating value of.....                                    | 82    |
| Electricity, generation of, by Diesel engines, cost of.....             | 121          | paraffin content of, effects of.....                     | 83    |
| Engine economy, factors effecting.....                                  | 102, 103     | prewarming of.....                                       | 63    |
| fuels, evaporation of.....                                              | 2            | specific gravity of, significance of.....                | 85    |
| heat balances of, figure showing.....                                   | 105          | storage of.....                                          | 63    |
| low-powered, requirements of.....                                       | 14           | sulphur content of, effects of.....                      | 83    |
| <i>See</i> Diesel engine, explosion oil-engine, Sabathé engine.         |              | viscosity of.....                                        | 84    |
| Exhaust gases, handling of.....                                         | 63           | water content of.....                                    | 82    |
| pipes, construction of.....                                             | 63           | Fuels for Diesel engine, acidity of.....                 | 87    |
| ports, position of.....                                                 | 35           | ash content of.....                                      | 88    |
| Explosion oil-engine, compression pressures in.....                     | 5            | boiling point of.....                                    | 87    |
| cost of fuel for.....                                                   | 17           | cost of, effect of.....                                  | 88    |
| cost of installing.....                                                 | 17           | flash point of.....                                      | 87    |
| cost of lubricant for.....                                              | 17           | heating value of.....                                    | 88    |
| definition of.....                                                      | 2            | impurities in.....                                       | 88    |
| description of.....                                                     | 3, 4         | specific gravity of.....                                 | 87    |
| explosion pressure in.....                                              | 5            | specifications for.....                                  | 87-89 |
| figure showing.....                                                     | 3, 4         | viscosity of.....                                        | 88    |
| fuel losses in.....                                                     | 16           | water contents of.....                                   | 88    |
| over-all economy of.....                                                | 15           | <i>See also</i> Diesel engines and oils named.           |       |
| temperatures in.....                                                    | 15           | Fuel requirements of Diesel engine.....                  | 96    |
| Explosion oil engines and Diesel engines, comparative economies of..... | 17           | Fuel valves for asphaltic oils.....                      | 87    |
| Explosion pressure in explosion oil engines..                           | 5            | Fulton Manufacturing Co., Diesel engine of, view of..... | 72    |
| F.                                                                      |              |                                                          |       |
| Filtering of fuel oil, need of.....                                     | 81           | G.                                                       |       |
| Flash point, desirable, of fuel.....                                    | 84           | Gas, coke-oven, heat cost of.....                        | 98    |
| determination of.....                                                   | 84           | Gas engines, advantages of.....                          | 106   |
| of fuel for Diesel engine.....                                          | 87           | efficiency of.....                                       | 98    |
| Flour mills, cost of power in with Diesel engines.....                  | 120          | operating cost of.....                                   | 98    |
| Flywheel, construction of.....                                          | 29           | Gas-producer plants, limitations in use of....           | 106   |
| limitations on size of.....                                             | 29           | Gasoline vapor, combustion of, air needed for.           | 86    |
| Flywheel factor, irregularity of.....                                   | 30, 31       |                                                          |       |
| Four-stroke cycle, advantages of.....                                   | 12           | H.                                                       |       |
| definition of.....                                                      | 2            | Heat balances of engines, figure showing.....            | 105   |
| oil engine, figure showing.....                                         | 3            | Heating value of fuel oil.....                           | 82    |
| Fractionation tests of petroleum, value of....                          | 80           | Horsepower, metric, definition of.....                   | 14    |
| Fuel, cost of, factors governing.....                                   | 114          | British, definition of.....                              | 14    |
| formula for computing.....                                              | 102          | Horsepower-hour, heat equivalent of.....                 | 102   |
| in relation to engine efficiency.....                                   | 104          | Hydrocarbons, combustion of.....                         | 77    |
|                                                                         |              | Hydrogen content, high, of fuel, advantage of.           | 84    |
|                                                                         |              | I.                                                       |       |
|                                                                         |              | Ice, cost of producing, with Diesel engine...            | 120   |
|                                                                         |              | K.                                                       |       |
|                                                                         |              | Kilowatt-hour, cost of generating, figure showing.....   | 112   |

| L.                                              |               | S.                                                |          |
|-------------------------------------------------|---------------|---------------------------------------------------|----------|
|                                                 | Page.         |                                                   | Page.    |
| Le Grand, Chas., cited.....                     | 76            | Sabathé engine, description of.....               | 11       |
| Lignite, average heat cost of.....              | 98            | definition of.....                                | 2        |
| Lignite-tar oils as Diesel engine fuel.....     | 89            | four-stroke cycle, pressure-volume dia-           |          |
| Liquid-fuel engines, types of.....              | 2             | gram of, figure showing.....                      | 12       |
| Locomobile engine, advantages of.....           | 104           | <i>See also</i> Diesel engine.                    |          |
| efficiency of.....                              | 98            | Scavenging in Diesel engines, method of.....      | 14       |
| operating cost of.....                          | 98            | Ships, Diesel engines for.....                    | 122      |
| Locomotives, Diesel engines for.....            | 121           | Southwark Foundry & Machinery Co., Diesel         |          |
| Low-speed engines, characteristics of.....      | 69            | engine of, view of.....                           | 76       |
| Lubricating pump, drive for, view of.....       | 26            | Specific gravity of fuel oil.....                 | 88       |
| Lubrication of Diesel engines.....              | 22, 94, 95    | significance of.....                              | 85       |
| Lunge, Georg, cited.....                        | 78            | Specifications for fuel oils.....                 | 87-89    |
| Lyons Atlas Co., Diesel engine of, view of....  | 70            | Stand-by losses in Diesel engines, absence of.... | 96       |
| <b>M.</b>                                       |               | Steam engine, efficiency of.....                  | 98       |
| McIntosh & Seymour Corporation, Diesel en-      |               | operating cost of.....                            | 98       |
| gine of, view of.....                           | 72            | use of exhaust of, for heating purposes....       | 107      |
| Main bearings, construction of.....             | 25            | Steam turbine, advantages of.....                 | 106, 116 |
| Mechanical efficiency of Diesel engine.....     | 65            | summary of.....                                   | 113      |
| Mexican crude oil as Diesel engine fuel.....    | 94            | comparative efficiency of, figure showing....     | 115      |
| Multicylinder engines, advantages of.....       | 29, 31        | cost of generating power with.....                | 111      |
| figure showing.....                             | 30            | efficiency of.....                                | 98       |
| <b>N.</b>                                       |               | operating cost of.....                            | 98       |
| National Transit Pump & Machine Co., Diesel     |               | Sulphur content of fuel oil, effects of.....      | 83       |
| engine of, view of.....                         | 74            | Sulphur dioxide, effect of on cylinder oil.....   | 84       |
| Natural gas, average heat cost of.....          | 98            | <b>T.</b>                                         |          |
| New London Ship & Engine Co., Diesel en-        |               | Tanks for fuel oil, construction of.....          | 63       |
| gine of.....                                    | 76            | Tar oil, ash content of.....                      | 90       |
| Nordberg Manufacturing Co., Diesel engine       |               | boiling point of.....                             | 90       |
| of, view of.....                                | 75            | burning point of.....                             | 90       |
| <b>O.</b>                                       |               | composition of.....                               | 91       |
| Oil engines, types of.....                      | 1             | flash point of.....                               | 90       |
| Oil-fuels, combustion of, factors affecting.... | 85            | for Diesel engine fuel.....                       | 91       |
| differences in.....                             | 93            | heating value of.....                             | 91       |
| <i>See</i> Fuel oil.                            |               | specific gravity of.....                          | 90       |
| Oil-gas tar, properties of.....                 | 93            | specifications for.....                           | 90       |
| <b>P.</b>                                       |               | suitability of, for engine fuel.....              | 78       |
| Palm oil, as engine fuel.....                   | 93            | use of.....                                       | 90       |
| Paraffin content of fuel oil, effects of.....   | 83            | <i>See also</i> Fuel oil.                         |          |
| Petroleum as engine fuel, requirements of....   | 79            | Tars, chamber retort, properties of.....          | 92       |
| tests of.....                                   | 79, 80        | coke-oven, properties of.....                     | 92       |
| <i>See also</i> Fuel oil.                       |               | horizontal retort, properties of.....             | 91       |
| Petroleum vapor, combustion of, air for.....    | 86            | oil-gas, properties of.....                       | 93       |
| Pistons, casting of.....                        | 27            | vertical retort, properties of.....               | 91       |
| cooling of.....                                 | 27            | water-gas, properties of.....                     | 93       |
| Diesel engine, construction of.....             | 26-28         | Two-stroke cycle, advantages of.....              | 12       |
| trunk, for Diesel engines, advantages of....    | 26            | definition of.....                                | 2        |
| Power, generation of, cost of, factors in.....  | 107, 118      | figure showing.....                               | 4        |
| figure showing.....                             | 108, 109, 110 | <b>V.</b>                                         |          |
| by Diesel engine.....                           | 110           | Valves of Diesel engine, figure showing.....      | 36       |
| Pressure-volume diagram for engine with four-   |               | Viscosity, desirable, of fuel oil.....            | 84, 88   |
| stroke cycle.....                               | 4             | of cylinder oil.....                              | 94, 95   |
| with two-stroke cycle.....                      | 5             | Volumetric efficiency of Diesel engine.....       | 67       |
| Prime movers, choice of.....                    | 114           | <b>W.</b>                                         |          |
| efficiency of, factors affecting.....           | 102           | Water for cooling, amount required.....           | 65       |
| influence of size on efficiency of, figure      |               | circulation of.....                               | 64       |
| showing.....                                    | 103           | supply of.....                                    | 64       |
| Pumping plants, prime movers for, efficiency    |               | Water-gas tar, properties of.....                 | 93       |
| of.....                                         | 120           | Water in fuel oil, effects of.....                | 82       |

